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RESEARCH MEMORANDUM

STUDY OF COMPRESSOR SYSTEMS FOR A GAS-GENERATOR ENGINE

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NATIONAL ADVISORY COMMITTEE
FOR AERONAUTICS

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STUDY OF COMPRESSOR SYSTEMS FOR A GAS-GENERATOR ENGINE

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SUMMARY

Various methods of providing compressor-capacity and pressure-ratio control in the gas-generator type of compound engine over a range of altitudes from sea level to 50,000 feet are presented.

The analytical results indicated that the best method of control is that in which the first stage of compression is carried out in a variable-speed supercharger driven by a hydraulic slip coupling. The second stage of compression could be either a rotary constant-pressure-ratio-type compressor or a piston-type compressor, both driven at constant speed. The analysis also indicated that the variation of the value of the load coefficient for the first and second stages of the rotary constant-pressure-type compressor combination was within reasonable limits and that the valve timing of the piston-type compressor could be kept constant for the range of altitudes covered. With respect to engine performance, other control schemes also appeared feasible. A variable-area turbine nozzle was shown to be unnecessary for cruising operation of the engine.

INTRODUCTION

An analysis of an aircraft-propulsion system known as a piston-type gas-generator engine is presented in reference 1. In this power plant a two-stroke-cycle, compression-ignition engine drives a compressor, which in turn supplies air to the engine. No other shaft work is abstracted from the engine. The gases from the generator comprising the compressor-engine combination are then utilized in a turbine, which produces the net useful work of the cycle. A diagrammatic sketch of this power plant is shown in figure 1.

The analysis of reference 1 indicates that such an engine may have low specific weight combined with low fuel consumption of the order of 0.32 pound per brake horsepower-hour, which has been confirmed, to a certain extent, by the experimental results reported

in reference 2. The reference analysis is idealized, however, in
that the problem of engine control was not considered. In order for
the performance of an actual engine to approach that of the ideal
curves, at least three steps must be accomplished: (1) A compres-
sor that is capable of operating over a range of air flows and
pressure ratios must be obtained; (2) similarly, a turbine that will
operate over the range of air flows and pressure ratios must be
obtained; and (3) satisfactory means of maintaining proper engine 1078
limits (peak cylinder pressure and turbine-inlet temperature) must
be evolved.

The first of the preceding steps, which pertains to the control of compressor capacity and compression ratio, is treated herein. The specific objective of this investigation, which was conducted at the NACA Lewis laboratory, is to evaluate various combinations of compressors and driving mechanisms with respect to engine performance as a function of altitude. Because it is currently impossible to make a complete evaluation with due consideration for such items as compressor weight, development problems, and control stability, the present analysis is based on the degeneration of the power output and the fuel economy of the gas-generator engine in question when compared with the ideal engine. Some qualitative discussion of compressor weight, however, is included.

METHOD OF ANALYSIS

The various compressor-control systems that were investigated included the following combinations of elements:

(1) Constant-pressure-ratio compressor with throttled inlet
(2) Multistage constant-pressure-ratio compressor with first stage (supercharging stage) driven by three-speed gear
(3) Multistage constant-pressure-ratio compressor with first stage (supercharging stage) driven by hydraulic slip coupling
(4) Constant-volume single-stage compressor
(5) Constant-volume compressor with throttled inlet
(6) Constant-volume compressor with means of varying volumetric capacity
(7) Supercharged constant-volume compressor comprising constant-pressure-ratio first stage (supercharging stage) driven by hydraulic slip coupling followed by final stage of compression in constant-volume compressor

In the preceding list, the terms "constant volume" and "constant pressure" refer to compressors exhibiting these characteristics at constant speed or effective speed. Thus the centrifugal compressor or the mixed-flow compressor would most nearly represent the constant-pressure-ratio group. The constant-volume class would include any of the positive-displacement compressors, such as the piston-type compressor or Roots blower. The axial-flow compressor would also fit into this class if suitable means were available for broadening its operating range so as to maintain high efficiency over a wide range of pressure ratios and air flows. Schematic diagrams of the various systems are shown in figure 2.

These possible combinations were derived by setting up the ideal requirements for the compressor for the gas-generator engine and then selecting the compressor systems most likely to fulfill these demands. The ideal requirements for the compressor (fig. 3) were computed from reference 1. Engine speed was not included as a control means because of the necessity for keeping the scavenge ratio within reasonable limits (reference 1). Thus, (a) if means were provided for keeping the scavenge ratio constant, such as a variable-area turbine nozzle, the reduction in engine speed necessary to decrease the compressor-pressure ratio to the required value as altitude is decreased (fig. 3) would result in greatly reduced air weight flow and hence reduced engine power output; and (b) if no means are provided for controlling scavenge ratio at will, as in the case of the fixed-area turbine nozzle, reduction in engine speed would result in increasing the scavenge ratio and hence burner mixture ratio beyond the usable range.

All the combinations were investigated with the analysis of reference 1 used as a basis. The changes required in the analysis as a result of the use of a specific compressor are given in the appendixes. All the combinations were analyzed with a fixed-area turbine nozzle and, in addition, systems 2 to 6 were also treated with a variable-area nozzle.

In analyzing the various combinations, the design conditions of the compressors were set to provide sufficient capacity and pressure ratio for engine operation at an altitude of 20,000 feet. At this altitude, the engine was assumed to operate with design operating limits of peak cylinder pressure of 1600 pounds per square inch, turbine-inlet temperature of 1800° F, and scavenge ratio of 1.0. For calculations of operation at higher altitudes, the turbine-inlet temperature and the compressor speed were held constant and the peak cylinder pressure was allowed to decrease. At lower altitudes, the generator-inlet pressure was so varied as to maintain both engine limits, unless the characteristics of the compressor system precluded this possibility, in which case the cylinder pressure was allowed to vary. When turbine-nozzle area was fixed, it was impossible to hold the scavenge ratio to a constant value of 1.0 at altitudes other than the design altitude because under choked conditions the area of the turbine nozzle determines the gas flow through the engine; however, the value of 1078 scavenge ratio did not vary greatly from 1.0 over the range of altitudes considered. When the area of the turbine nozzle was made variable, the scavenge ratio was kept constant at a value of 1.0. The scavenge ratio of the ideal gas-generator engine was 1.0 for all altitudes.

Systems 2, 3, and 7, which utilize a rotary supercharging compressor, were so arranged that the supercharger operated at rated speed at an altitude of 20,000 feet and idled at sea level. In the system involving the three-speed gear combination, the supercharger was assumed to idle at a pressure ratio of 1 at altitudes below 10,000 feet and to operate in the low-speed gear ratio at altitudes between 10,000 and 20,000 feet. Above 20,000 feet, the supercharger operated in high speed.

The over-all efficiency of any combination of compressors was made equal to 0.85 at an altitude of 20,000 feet. Although this practice resulted in rather high stage efficiencies, it was necessary that the data agree with that of reference 1 at this basic condition. The lack of accurate data on the efficiencies of constant-volume compressors, as, for example, that of the axial-flow compressor at off-design conditions or the piston-type compressor at high piston speeds, precluded comparison of the constant-pressure and constant-volume compressors on an efficiency basis. The most reliable NACA data on efficiency of piston-type compressors, however, indicate values in the range from 0.85 to 0.95, which is entirely compatible with the general assumption. No changes in efficiency with changes in the specific flow for the constant-pressure compressors were considered; instead, the pressure ratios of these compressors were limited to values that would permit a moderate operating range. This limitation was necessary in order to avoid considerations of compressor design, which are beyond the scope of this report.

RESULTS AND DISCUSSION

Because of the inherent differences in compressor characteristics, the results of the analyses of constant-pressure-ratio compressors and constant-volume compressors are discussed separately. The effects of fixing the turbine-nozzle area and of designing for high altitudes are also considered.

Constant-Pressure-Ratio Compressor

Effect of throttling. - The effect of throttling the compressor inlet as a control means is shown in figure 4, which shows that throttling provides a means of maintaining the engine limits (peak burner pressure and turbine-inlet temperature) in the altitude range up to 20,000 feet, but that the engine is penalized by large reductions in brake output. This penalty is a natural result of the high compressor power load in the gas-generator type of engine. Consequently, although throttling is considered to be applicable to the gas-generator engine with the constant-pressure-ratio compressor, it is undesirable from a standpoint of low-altitude power output.

Effect of three-speed supercharging-compressor drive. - An inspection of figure 5 indicates that the performance of a gas-generator engine with a three-speed supercharging compressor is subject to large power losses at altitudes just below those at which the gear changes take place. This fact, coupled with the difficulty of providing a change gear and a clutch capable of handling the required powers, indicates that this system is undesirable for gas-generator application.

Effect of hydraulic-slip-coupling supercharging-compressor drive. - The performance of the gas-generator engine when equipped with a multistage constant-pressure-ratio compressor comprising a supercharging compressor driven by a hydraulic slip coupling and followed by a constant-speed final compression stage is presented in figure 6. It will be noted that only a small loss in power between sea level and 20,000 feet occurs when the coupling is used. The efficiency of the coupling is a linear function of the slip; it is equal to zero at 100-percent slip and approaches 100 percent as the slip approaches zero. Because the compressor torque varies with the square of the speed, it can be shown that the coupling power loss is a maximum at 50-percent slip, and is equal to one-fourth of the full-load compressor power if the air flow remains constant. Furthermore, the compressor driven by the coupling comprises a fraction of the total compressor in the gas-generator engine. The small coupling-power loss relative to the total compressor load indicates a logical reason for the small influence of the coupling on the over-all performance of the gas-generator engine.

Constant-Volume Compressor

In the case of the constant-volume compressor, the piston-type compressor, in particular, has numerous inherent unique advantages for gas-generator-engine applications, some of which may be listed as follows:

(a) Provides a compact, light machine for operation at the low air flows and high pressure ratios required by the gas-generator engine. 1078

(b) Possesses a broader operating range (high efficiency over wide range of pressure ratios and air flows) than equivalent rotary-compressor types.

(c) Permits a higher compressor efficiency to be obtained at high pressure ratios.

(d) Delivers a positive supply of air under all operating conditions, including starting and idling.

Advantage (a) deserves some elaboration. The same piston-type compressor is capable of operating over an extremely wide range of pressure ratios, for example, from 2 up to values of 20 or 25. At the same time, the weight is fixed by structural stiffness requirements, so that little if any change in weight accompanies a change in pressure ratio. In the equivalent constant-pressure-ratio compressor, an increase in pressure ratio can be obtained only by staging with a consequent increase in weight. Thus, as the required pressure ratio is increased, the piston-type compressor becomes lighter relative to the constant-pressure compressor.

Also, because of the staging required to obtain high pressure ratios, the efficiency of each stage of the constant-pressure compressor must be extremely high in order that the over-all efficiency of the constant-pressure-compressor combination may approach the efficiency easily attainable with the constant-volume piston-type compressor (on the order of 0.85). The fact that such efficiencies may not be obtainable with the constant-pressure compressor increases the desirability of using the piston-type compressor.

The prime advantage of the constant-pressure compressor is, of course, its extremely high volume-flow capacity, leading to a low specific weight. At low volumetric flow rates, however, this advantage disappears to a certain extent because of the difficulty of designing these compressors with small flow passages and clearances and with high rotational speeds.

Consequently, the ideal circumstances for the use of the piston-type compressor are low volume-flow rates and high pressure ratios. These circumstances are present in the gas-generator engine, particularly in the final stages of compression.

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The size of the piston-type compressor need not be excessive. The ratio of compressor volume to burner volume is 10.35 for the case of the gas-generator engine with the unsupercharged piston-type compressor and the fixed-area turbine nozzle at 20,000 feet. If the compressor is supercharged for all altitudes other than sea level, this ratio becomes 6.01. If supercharging is used also for sea-level operation, however, the compressor-volume – burner-volume ratio may be made to approach unity. Inlet ram due to flight speed further reduces the required volume ratio. Furthermore, fitting the required compressor volume into the gas-generator engine and furnishing the necessary reciprocating motion is not a great design problem. In certain engine configurations, such as the axial engine, the reciprocating motion is readily available and a compact, small-frontal-area engine may be easily attained. Further decrease in size of the piston-type compressor may be obtained by making the compressor double-acting.

One disadvantage may be ascribed to the piston-type compressor in that considerable work will be required to develop it into a practical high-speed machine; however, this same disadvantage applies to rotary compressors required to operate at very high pressure ratios and efficiencies.

From these practical considerations, the piston-type compressor seemed an attractive choice for a constant-volume-type compressor for use with the piston-type gas-generator engine, although the results would be applicable to other forms of constant-volume compressors. For these reasons, the piston-type compressor was included in the analysis.

Effect of fixed-displacement constant-volume compressor. - Figure 7 shows the performance of a gas generator using a fixed-displacement piston-type compressor (that is, one equipped with automatic compressor valves), as compared with the performance of the ideal gas-generator engine. With this type of compressor, rate of air flow through the engine is substantially dependent upon only compressor speed. Consequently, with a fixed restriction in the turbine, turbine-inlet pressure must increase until the flow through the turbine matches that through the compressor. The resultant high manifold pressures and compressor loads cause the

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burner pressure to increase above the limiting value at altitudes below the 20,000-foot critical altitude; therefore, the system is not usable.

Effect of throttled fixed-displacement constant-volume compressor. - Throttling the exhaust from the fixed-displacement piston-type compressor will only make the situation regarding burner pressure worse inasmuch as such a change will increase the compressor load without appreciably affecting the manifold pressure. Throttling the inlet to this compressor, however, permits the air flow and consequently the manifold pressure to be reduced to a point at which limiting values of burner pressure are attained at altitudes below the critical altitude. Actually, the compressor load is higher than that of the ideal engine, so the manifold pressure must be lower than that of the ideal engine, causing a reduction in performance. Figure 8 shows the performance of such a throttled engine. The large loss in brake output and the increase in fuel consumption between 20,000 feet and sea level makes this system unattractive for gas-generator use.

Effect of variable-displacement constant-volume compressor. - The performance of a gas-generator engine equipped with a variable-displacement compressor, that is, a piston-type compressor equipped with mechanically actuated valves that permit variable timing with this device, is illustrated in figure 9. On the basis of engine performance, this system is quite satisfactory. The piston compressor, however, must handle air at ambient atmospheric conditions. Under this circumstance of high volumetric air flow, the piston compressor may become relatively heavy as compared with equivalent rotary types. This fact, in addition to the valve complication, makes this scheme undesirable for gas-generator use.

Effect of supercharged fixed-displacement constant-volume compressor. - The use of a variable-speed supercharger driven by a hydraulic slip coupling with a fixed-displacement piston-type compressor affords a means of adjusting air flow through the engine and utilizing all the piston-type-compressor displacement at all altitudes. The performance of such a combination is presented in figure 10. The curves of this figure indicate that this scheme is the most promising of those incorporating a piston-type compressor. It is interesting to note that, when a supercharging compressor is used, the time at which the reciprocating-compressor valves open and close is substantially constant with changes in altitude (fig. 11), so that mechanical valves with fixed timing can be substituted for the automatic valves without incurring a penalty in engine performance.

Comparison of Piston-Type and Rotary Compressors

The two most successful methods of satisfying the compressor requirements in the gas-generator engine appear to be the use of a variable-speed supercharging compressor followed by either a constant-volume piston-type or a constant-pressure stage for compression of the air to burner-inlet pressure. Unfortunately, the lack of data concerning the piston-type-compressor weights and efficiencies precludes a definite evaluation of the two compressor types. Figure 12 shows the performance of the two gas-generator engines as calculated by the methods of this report. Below the critical altitude, the performance of the combination with the piston-type constant-volume compressor is slightly superior to that obtained with the constant-pressure combination. The slight difference in performance between the two systems is caused by the fact that the pressure ratio across the constant-pressure compressors is a function of the inlet temperature, whereas that for the reciprocating compressor is constant.

The variations of values of load coefficient Q/n for the first stage (supercharger) and second stage as a function of altitude for the multistage rotary constant-pressure compressor with first stage driven by a hydraulic slip coupling is shown in figure 13. Although the values of Q/n vary widely for the supercharger below an altitude of 10,000 feet, the tip speed of this compressor is quite low, which may keep it out of a surging condition. The variation of Q/n for the second stage is within present acceptable limits. The performance of the supercharger when used with the piston-type compressor is substantially the same as that shown in figure 13.

Effect of Variable-Area Turbine Nozzle

Examination of the data comparing the fixed-area nozzle with the variable-area nozzle (figs. 14 to 18) indicates that only a slight reduction in performance is incurred through the use of the fixed-area nozzle. Generally, a small drop in power occurs at altitudes below the 20,000-foot critical altitude, which is caused by a change in scavenge ratio incurred through lack of adequate air-flow control. In general, the power loss is small in comparison with the problems incurred in the successful development of such a device; therefore, the variable-area turbine nozzle will not be further considered in this report.

Effect of Designing for High Altitudes

The performance curve for the gas-generator engine equipped with a variable-speed supercharger driven by a hydraulic coupling (fig. 6) indicates that some losses are incurred in going from sea level to the design altitude. Because these losses increase in magnitude as the design altitude is raised, it is of interest to examine the case in which the engine is designed for a very high altitude. In figure 19, the performance of the gas generator with the variable-speed supercharging compressor is shown when the optimum altitude is 40,000 feet. It is noted that in this case, the loss in brake output at moderate altitudes (0 to 20,000 ft) is more serious than for the case of the engine designed for 20,000-foot optimum altitude. More brake output, however, is available for climbing to an altitude of 40,000 feet and above, and this engine should therefore be more satisfactory in applications where a high-altitude engine is warranted.

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SUMMARY OF RESULTS

The results of the analysis presented herein for various methods of providing compressor-capacity and pressure-ratio control for the gas-generator engine operating over a range of altitudes with constant peak cylinder pressure and constant turbine-inlet temperature may be summarized as follows:

1. The best method of compressor control appeared to be that in which the first stage of compression consisted of a variable-speed supercharger that was driven by a hydraulic slip coupling. The second stage of compression could be either a rotary constant-pressure-ratio-type compressor or a piston-type compressor, both driven at constant speed. The variation of load coefficient Q/n for the first and second stages of compression when a constant-pressure-ratio final-compression stage was used remained within reasonable limits over the altitude range considered. With a constant-volume compressor for final compression, the valve timing of the piston-type compressor could be held constant over the altitude range considered.

2. Other control methods, which appeared feasible with regard to engine performance, are the use of a constant-volume, piston-type compressor with variable valve timing or a constant-pressure compressor, the first stage of which is driven by a three-speed gear.

3. Throttling generally produced large power losses at other than the design altitude in the gas-generator engine.

4. For cruising operation of the engine, the complication of a variable-area turbine nozzle was not warranted.

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APPENDIX A

SYMBOLS

The following symbols are used in this report:

C	percentage clearance in piston-type compressor
$c_{p,a}$	specific heat at constant pressure of compressor air, 0.243 Btu/(lb)(°R)
$c_{p,g}$	specific heat at constant pressure of turbine gases, 0.270 Btu/(lb)(°R)
g	acceleration due to gravity, 32.2 ft/sec^2
h_c	lower heat of combustion of fuel, 18,500 Btu/lb fuel
J	mechanical equivalent of heat, ft-lb/Btu
N	engine speed, cycles/sec
P	total pressure, lb/sq in. absolute
P_e	burner-exhaust pressure, lb/sq in. absolute
P_m	burner-inlet manifold pressure, lb/sq in. absolute
p	static pressure, lb/sq in. absolute
p_a	ambient air pressure, lb/sq in.
p_c	burner compression pressure, lb/sq in. absolute
Q/n	load coefficient, cu ft/revolution
q_{ad}	pressure coefficient of compressor
R_e	expansion ratio of fluid in burner
R_m	over-all mixture ratio, lb fuel/lb air
$R_{m,b}$	mixture ratio in burner, lb fuel/lb air
R_p	pressure ratio in piston-type compressor

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$R_{p,1}$ pressure ratio in first stage of compression, rotary compressor

$R_{p,2}$ pressure ratio in second stage of compression, rotary compressor

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$R_{p,o}$ over-all pressure ratio across compressors

R_s scavenge ratio (ratio of volume of air flowing through burner per cycle measured at burner-inlet conditions to volume of burner)

T temperature, °R

T_a ambient air temperature, °R

T_c burner compression temperature, °R

T_g mean turbine-inlet temperature, °R

T_m burner-inlet temperature, °R

T_s temperature in burner at end of scavenging, °R

v volume, cu ft

v_b total volume of burner above ports, cu ft

v_c total volume of reciprocating compressor, cu ft

W_c work of compressor, Btu/lb air

W_t work of turbine, Btu/lb air

γ ratio of specific heats of turbine gases

η_{ad} adiabatic over-all efficiency of rotary compressor

η_b burner brake thermal efficiency (actual)

η_c adiabatic efficiency of piston-type compressor

$\eta_{c,1}$ adiabatic efficiency of rotary compressor, first stage

$\eta_{c,2}$ adiabatic efficiency of rotary compressor, second stage

$\eta_{c,o}$ over-all adiabatic efficiency of compressor unit

η_r reduction-gear efficiency, 0.95

η_s scavenging efficiency (ratio of volume of air remaining in burner at end of scavenging process, measured at inlet conditions, to volume of burner) 1078

η_{sl} efficiency of slip coupling

η_t adiabatic turbine efficiency, total to static, 0.85

ρ density, lb/cu ft

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APPENDIX B

ANALYSIS OF CYCLE

In general, the analysis of the cycle is similar to that of reference 1. A condensation of that analysis is given here for convenience. Because reference 1 is an idealized analysis, certain variations must be made when considering the gas-generator engine with regard to control. These specific details will be presented in the succeeding appendixes.

The following method of analysis was used to estimate the idealized performance of the gas-generator engine.

Compressor calculations. - The over-all performance of the compressor unit on the basis of work done per pound of air handled is

$$W_c = c_{p,a} \left(T_m - T_a\right) \tag{B1}$$

where

$$T_m = \frac{T_a}{\eta_{c,o}} \left(R_{p,o}^{\;0.283} - 1\right) + T_a \tag{B2}$$

Scavenge efficiency and scavenge ratio. - The scavenge ratio of the piston-type burner is given by the equation

$$R_s = 0.0910 \sqrt{\left(1 - P_e/P_m\right) T_m} \tag{B3}$$

and the scavenge efficiency by the equation

$$\eta_s = 1 - e^{-R_s}$$

The temperature of the gases in the cylinder at the end of the scavenge process is

$$T_s = \frac{T_m}{1 - \left(1 - \frac{T_m}{2000}\right) e^{-R_s}} \tag{B4}$$

It is assumed that the inlet and exhaust manifolds are sufficiently large so that total and static pressures are approximately equal.

Burner efficiency. - The brake thermal efficiency of the piston-type burner is

$$\eta_b = 0.925 - \left(\frac{1}{R_e}\right)^n \tag{B5}$$

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where

$$n = 0.3867 - \frac{6.5}{\frac{6.65}{R_{m,b}} - 35} - \frac{0.043}{R_e} \tag{B6}$$

The relation between the work of the compressor and the burner output is given by the equation

$$W_c = \eta_b R_m h_c \tag{B7}$$

Because no simple relation between η_b and R_m exists, a trial-and-error method of solution is necessary.

The burner efficiency was first approximated by the equation

$$\eta_b' = 0.925 - \left(\frac{1}{R_e}\right)^{0.32} \tag{B8}$$

(The use of the prime on the symbols indicates an approximation.) With this value, the over-all fuel-air ratio was estimated by

$$R_m' = \frac{W_c}{\eta_b' \, h_c} \tag{B9}$$

This value of over-all mixture ratio was modified to represent approximately the mixture ratio existing in the cylinder by use of the equation

$$R_{m,b}' = \frac{R_s \, R_m'}{\eta_s} \tag{B10}$$

With this value of $R_{m,b}'$, the burner-efficiency calculations were repeated and the corrected mixture ratios were found from equations (B7) and (B10).

Maximum burner pressure. - Curves of the ratio of ideal peak burner pressure to compression pressure as a function of mixture ratio and with compression temperature as a parameter were prepared by use of the fuel-air cycles and methods of reference 3 for the rich mixtures and by use of air cycles for very lean mixtures. These pressure ratios were modified by the factor

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$$F = -3.75\ R_{m,b} + 1.0 \tag{B11}$$

to bring the ideal ratios into accordance with engine data.

Compression pressure and temperature were computed from the equations

$$p_c = P_e\ R_e^{1.35} \tag{B12}$$

$$T_c = T_s\ R_e^{0.35} \tag{B13}$$

Turbine-inlet temperature. - A heat balance applied to the gas-generator engine showed that, with the assumption of a heat loss equivalent to 18 percent of the fuel-heat input, the turbine-inlet temperature was given by the equation

$$T_g = \frac{(1\text{-}0.18)\ h_c R_m + c_{p,g} T_a\ (1+R_m)}{c_{p,g}\ (1+R_m)} \tag{B14}$$

Turbine power. - The output of the turbine in Btu per pound of air is

$$W_t = \eta_t\ c_{p,g}\ T_g\ (1+R_m)\ \left[1 - \left(\frac{p_a}{P_e}\right)^{\frac{\gamma-1}{\gamma}}\right] \tag{B15}$$

where

$$\gamma = 1.34$$

and

$$c_{p,g} = 0.27$$

Unit performance calculations. - The output of the gas-generator engine on the basis of Btu per cycle per cubic inch of burner volume is

$$\text{Brake output} = \frac{\eta_r W_t R_s}{1728} \frac{144\ P_m}{53.3 T_m} \tag{B16}$$

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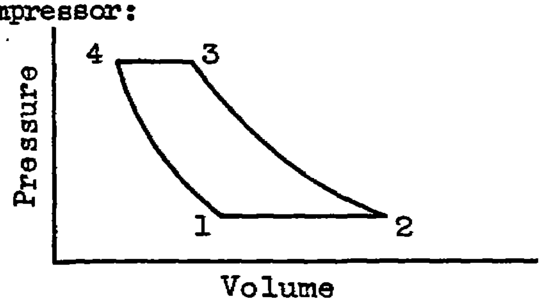
and the brake specific fuel consumption is

$$\text{bsfc} = \frac{2545\ R_m}{\eta_r W_t} \tag{B17}$$

APPENDIX C

PISTON-TYPE COMPRESSOR

The pressure ratio across the piston-type compressor uniquely determines the scavenge ratio of the piston-type burner, as may be shown by the following compressor-cycle analysis, which uses as a basis the idealized pressure-volume indicator diagram of a piston-type compressor:



Inasmuch as the temperature rise across the compressor is based on the adiabatic efficiency, for the ratio of specific heats of 1.395

$$\frac{T_3}{T_2} = 1 + \frac{1}{\eta_c}\left(R_p^{\;0.283} - 1\right) \tag{C1}$$

If the percentage of clearance C is defined as

$$C = \frac{v_4}{v_2 - v_4}$$

where $v_2 - v_4$ is the compressor displacement, then

$$\frac{v_4}{v_2} = \frac{C}{1 + C}$$

The weight of air delivered by the compressor per cycle is

$$(v_3 - v_4)\,\rho_3$$

and the scavenge ratio R_s is

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$$R_s = \frac{v_3 - v_4}{v_b\ \rho_3}\rho_3 = \frac{v_3 - v_4}{v_b}$$

when it is assumed that both the burner and compressor operate at the same number of cycles per unit time. Because

$$\frac{p_4 v_4}{T_4} = \frac{p_1 v_1}{T_1}$$

and

$$\frac{p_3 v_3}{T_3} = \frac{p_2 v_2}{T_2}$$

then

$$v_3 - v_4 = v_2\left(\frac{p_2}{p_3}\frac{T_3}{T_2} - \frac{v_4}{v_2}\right)$$

and

$$v_3 - v_4 = v_2\left(\frac{1}{R_p}\frac{T_3}{T_2} - \frac{C}{1 + C}\right)$$

so that

$$R_s = \frac{v_2}{v_b}\left(\frac{1}{R_p}\frac{T_3}{T_2} - \frac{C}{1 + C}\right)$$

and

$$R_s = \frac{v_c}{v_b}\left\{\frac{1}{R_p}\left[1 + \frac{1}{\eta_c}\left(R_p^{0.283} - 1\right)\right] - \frac{C}{1 + C}\right\} \qquad \text{(C2)}$$

The value of C in this analysis is 0.03. The volume ratio v_c/v_b is determined from the limiting conditions of engine operation at an altitude of 20,000 feet and a scavenge ratio of 1.0. For an unsupercharged compressor, the value of this ratio is 10.35.

<u>Variable-displacement compressor.</u> - The displacement of the piston-type compressor may be effectively varied by controlling the valve timing by means of some mechanical arrangement. It is possible to decrease the displacement by: (1) closing the inlet valve early, (2) closing the inlet valve late, or (3) closing the exhaust valve late. Method (3) is the one considered in this analysis, and if X is the percentage of the piston stroke that the piston has returned when the exhaust valve is closed late, the scavenge ratio will then be

$$R_s = \frac{v_c}{v_b}\left\{\frac{1}{R_p}\left[1 + \frac{1}{\eta_c}\left(R_p^{\,0.283} - 1\right)\right] - \frac{X + C}{1 + C}\right\} \tag{C3}$$

APPENDIX D

HYDRAULIC SLIP COUPLING

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The hydraulic slip coupling is perhaps the simplest and most practical means of obtaining changes in the speed of the first-stage compressor. In addition, the hydraulic coupling provides some shock and vibration isolation. Reference 4 lists some of the design features of current slip couplings.

The size of the coupling needed to transmit the power to the first-stage compressor need not be excessive because the torque and the horsepower of the fluid coupling varies as the fifth power of the diameter and only a relatively small change in diameter will be necessary to cover a large range in transmitted power. Current fluid couplings can therefore be used in the gas-generator engine with only a small variation in the size of the coupling unit.

The type of hydraulic coupling considered in this analysis is the scoop-control hydraulic coupling. Variations in speed of the secondary are made possible by means of an adjustable-scoop tube, extending from the impeller section to a rotating reservoir. Oil passes from the working circuit through the nozzles provided in the inner casing and collects in the rotating reservoir from which it is returned to the coupling circuit. The scoop tube is mounted off center, so that in its fully extended position it handles all the oil in the rotating reservoir, but when brought to a fully retracted position, all the oil drains into the reservoir and the coupling is fully disconnected. Varying the scoop position from a fully extended to a fully retracted position varies the speed of the secondary unit from maximum to zero values.

In the type of fluid coupling considered, in which there is no torque-reaction member, the torque input equals the torque output, and the efficiency is equal to the ratio of secondary to primary speed, so that

$$\frac{\text{hp out}}{\text{hp in}} = \frac{N_s}{N_p} = \eta_{s,l} \qquad \text{(D1)}$$

where

N_p primary speed, rps

N_s secondary speed, rps

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APPENDIX E

ROTARY COMPRESSOR

The rotary compressor in this analysis is considered to be either a centrifugal or a mixed-flow compressor. Its performance characteristics are assumed to follow the same laws as those under which a centrifugal compressor operates.

The temperature rise across each stage is therefore

$$\Delta T = \frac{T_1}{\eta_{ad}}\left[\left(\frac{P_2}{P_1}\right)^{0.283} - 1\right] \tag{E1}$$

where η_{ad} is the stage efficiency and P_2/P_1 is the pressure ratio across the stage. The work required in Btu per pound of air is

$$W_c = c_{p,a}\,\Delta T \tag{E2}$$

When speed changes and their effects on the performance of the compressor are considered, it is convenient to use the forms involving the pressure coefficient

$$q_{ad} = \frac{Jg\ c_{p,a}\ T_1\left[\left(\frac{P_2}{P_1}\right)^{0.283} - 1\right]}{V_T^2} \tag{E3}$$

where V_T is the tip speed in feet per second. Because

$$W_c = \frac{J\ c_{p,a}\ T_1}{\eta_{ad}}\left[\left(\frac{P_2}{P_1}\right)^{0.283} - 1\right] \tag{E4}$$

then

$$W_c = \frac{q_{ad}}{\eta_{ad}}\frac{V_T^2}{g} \tag{E5}$$

and

$$\frac{P_2}{P_1} = \left(1 + \frac{q_{ad}\ V_T^2}{Jg\ c_{p,a}\ T_1}\right)^{3.535} \tag{E6}$$

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Rotary compressor with slip coupling. - If the value K represents the speed ratio of the slip coupling

$$K = \frac{N_s}{N_p} = \eta_{sl} \tag{E7}$$

then the work put into the primary side of the slip coupling is

$$W_c = \frac{q_{ad}}{\eta_{ad}} \frac{V_T^2}{g} \frac{1}{K} \tag{E8}$$

and if

$$V_T = N_s D$$

where D is the diameter of the compressor

$$W_c = \frac{q_{ad}}{\eta_{ad}} \frac{N_s^2 D^2}{g} \frac{N_p}{N_s}$$

so that

$$W_c = \frac{q_{ad}}{\eta_{ad}} \frac{N_p D^2}{g} N_s$$

and inasmuch as

$$N_s = K\ N_p$$

$$W_c = \frac{q_{ad}}{\eta_{ad}} \frac{N_p^2 D^2 K}{g} \tag{E9}$$

For a given compressor N_p and D are constants and are selected after determination of the operating range required.

The pressure ratio is given by the equation

$$\frac{P_2}{P_1} = \left(1 + \frac{q_{ad}\ D^2 N_p{}^2\ K^2}{Jg\ c_{p,a}\ T_1}\right)^{3.535} \tag{E10}$$

When in this analysis a two-stage compressor is used with the first stage driven at a variable speed, the value of K^2 necessary to obtain a desired over-all pressure ratio across the compressors may be found by the following procedure:

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$$R_{p,1} = \left(1 + K^2 \frac{A}{T_1}\right)^{3.535}$$

where the constant A takes into account the diameter of the first-stage compressor, the speed of the primary member of the coupling, the pressure coefficient, and Jgc_p. Then,

$$T_2 = \frac{T_1}{\eta_{c,1}}\left(R_{p,1}{}^{0.283} - 1 + \eta_{c,1}\right)$$

where T_2 is the temperature of the air leaving the first stage, and

$$R_{p,2} = \left(1 + \frac{B}{T_2}\right)^{3.535}$$

where B is a constant that takes into account the diameter of the second-stage compressor, its speed, the pressure coefficient, and Jgc_p. Now

$$K^2 = \frac{T_1}{A}\left(R_{p,1}{}^{0.283} - 1\right)$$

and

$$R_{p,1} = \frac{R_{p,o}}{R_{p,2}}$$

so that

$$K^2 = \frac{T_1}{A}\left(\frac{R_{p,o}{}^{0.283}}{1 + \frac{B}{T_2}} - 1\right)$$

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$$K^2 = \frac{T_1}{A}\left[\frac{R_{p,o}{}^{0.283}}{1 + \frac{B}{T_1}\,\frac{\eta_{c,1}}{\left(R_{p,1}{}^{0.283} - 1 + \eta_{c,1}\right)}} - 1\right]$$

and

$$K^2 = \frac{T_1}{A}\left[\frac{R_{p,o}{}^{0.283}}{1 + \frac{B}{T_1}\left(\frac{\eta_{c,1}}{K^2\,\frac{A}{T_1} + \eta_{c,1}}\right)} - 1\right] \tag{E11}$$

This equation may be transposed into a quadratic equation in K^2 from which K^2 may easily be found.

Compressors in series. - When two compressors are connected in series, the over-all efficiency is different from the stage efficiencies. Because comparison of the performance in this analysis is to be made with that of reference 1, it is necessary to know the relation between stage and over-all efficiency in order that the over-all compressor efficiency in this analysis at an altitude of 20,000 feet may equal the compressor efficiency at 20,000 feet given in reference 1.

The over-all adiabatic efficiency is given by the equation

$$\eta_{c,o} = \frac{R_{p,o}{}^{0.283} - 1}{\frac{1}{\eta_{c,1}}\left(R_{p,1}{}^{0.283} - 1\right) + \frac{1}{\eta_{c,2}}\left[\frac{1}{\eta_{c,1}}\left(R_{p,1}{}^{0.283} - 1\right) + 1\right]\left(R_{p,2}{}^{0.283} - 1\right)} \tag{E12}$$

where

$$R_{p,1}\,R_{p,2} = R_{p,o} \tag{E13}$$

APPENDIX F

TURBINE NOZZLE

The mass flow through a convergent nozzle with critical flow is

$$W = \sqrt{2g}\,\frac{P_e \times 144}{\sqrt{T_g}}\,\frac{A}{\sqrt{R}}\sqrt{\frac{\gamma}{\gamma+1}\left(\frac{2}{\gamma+1}\right)^{\frac{2}{\gamma-1}}} \tag{F1}$$

where

W weight flow, lb/sec

A area, sq ft

R gas constant, ft-lb/(lb)(^{o}R)

if

$$\gamma = 1.34$$

and

$$R = 53.35$$

then

$$W = 75.35\,A\,\frac{P_e}{\sqrt{T_g}} \tag{F2}$$

This mass flow must be equal to the sum of the air flow through the burner and the fuel flow. Thus

$$144\,R_s\,\frac{P_m}{T_m}\,\frac{Nv_b}{R} = \frac{75.35\,A}{1 + R_m}\,\frac{P_e}{\sqrt{T_g}}$$

so that

$$\frac{P_e}{P_m} = R_s\,\frac{2.704}{75.35}\,\frac{1}{\theta}\,\frac{\sqrt{T_g}}{T_m}\,(1 + R_m) \quad \text{where} \quad \theta = \frac{A}{Nv_b}$$

but

$$\frac{P_e}{P_m} = 1 - \left(\frac{1}{T_m}\right)\left(\frac{R_s}{0.0910}\right)^2$$

so that if 1078

$$T_g = 2260^\circ \text{ R}$$

$$\theta = \frac{(1 + R_m)}{0.5862\left(\frac{T_m}{R_s} - 120.8\ R_s\right)} \tag{F3}$$

and

$$R_s = \frac{1}{100}\left[\sqrt{\frac{0.4990\ (1 + R_m)^2}{\theta^2} + 82.8\ T_m} - \frac{0.7065\ (1 + R_m)}{\theta}\right] \tag{F4}$$

For fixed turbine-nozzle-area operation, the value of turbine-nozzle area is fixed at that required for operation at an altitude of 20,000 feet at a scavenge ratio of 1.0, a peak burner pressure of 1600 pounds per square inch absolute, and a turbine-inlet pressure of 2260° R. This value of turbine-nozzle area θ was 0.001686 square foot per cubic foot of burner volume per cycle per second.

When a fixed-turbine-nozzle area is used in conjunction with the reciprocating compressor, it is necessary that the operating conditions, for which the scavenge ratio determined by the reciprocating compressor equals the scavenge ratio determined by the turbine nozzle, be obtained by means of a graphical solution. Other graphical solutions are, of course, necessary even if the turbine-nozzle area is not fixed as there is no convenient expression relating burner-inlet pressure, burner-expansion ratio, and mixture ratio to the limiting conditions of peak burner pressure and turbine-inlet temperature.

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3. Hersey, R. L., Eberhardt, J. E., and Hottel, H. C.: Thermodynamic Properties of the Working Fluid in Internal-Combustion Engines. SAE Jour. (Trans.), vol. 39, no. 4, Oct. 1936, pp. 409-424.

4. Alison, N. L.: Fluid Transmission of Power. SAE Jour. (Trans.), vol. 48, no. 1, Jan. 1941, pp. 1-8.

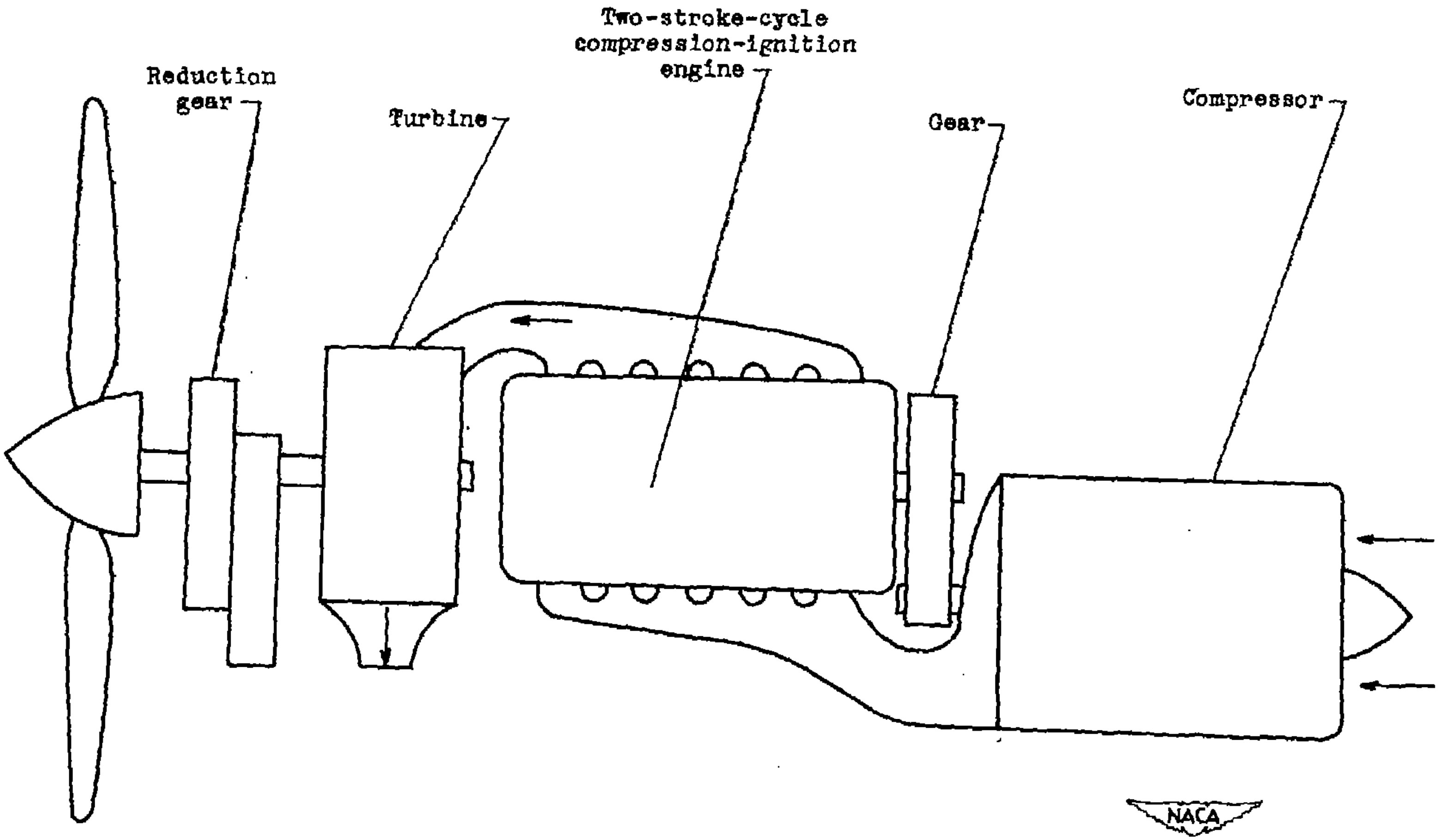
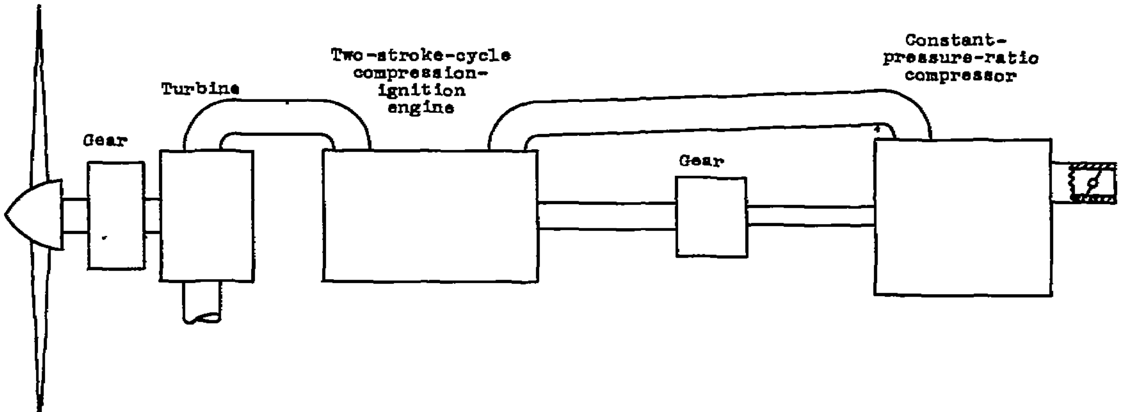
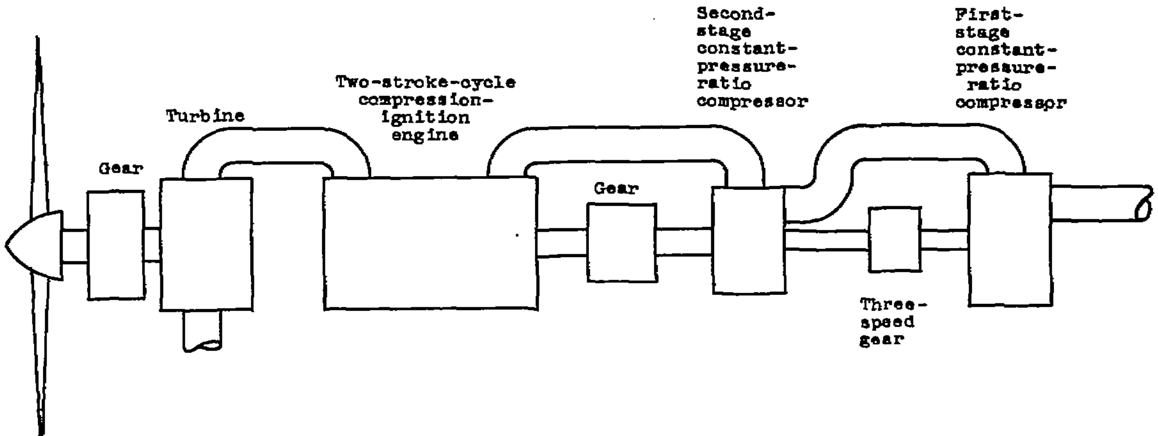


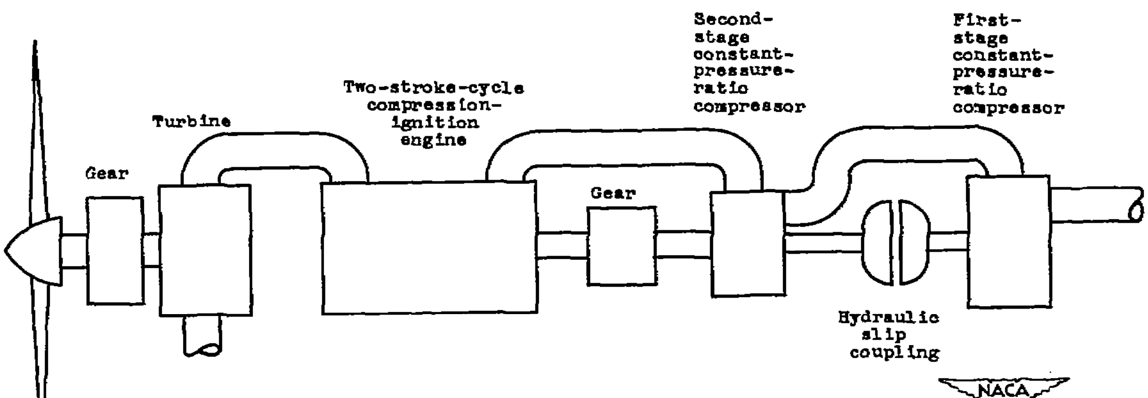
Figure 1. - Diagrammatic sketch of gas-generator engine used in analysis (reference 1).



(a) Constant-pressure-ratio compressor with throttled inlet.



(b) Multistage constant-pressure-ratio compressor with first stage driven by three-speed gear.



(c) Multistage constant-pressure-ratio compressor with first stage driven by hydraulic slip coupling.

Figure 2. Schematic diagrams of systems used in analysis.

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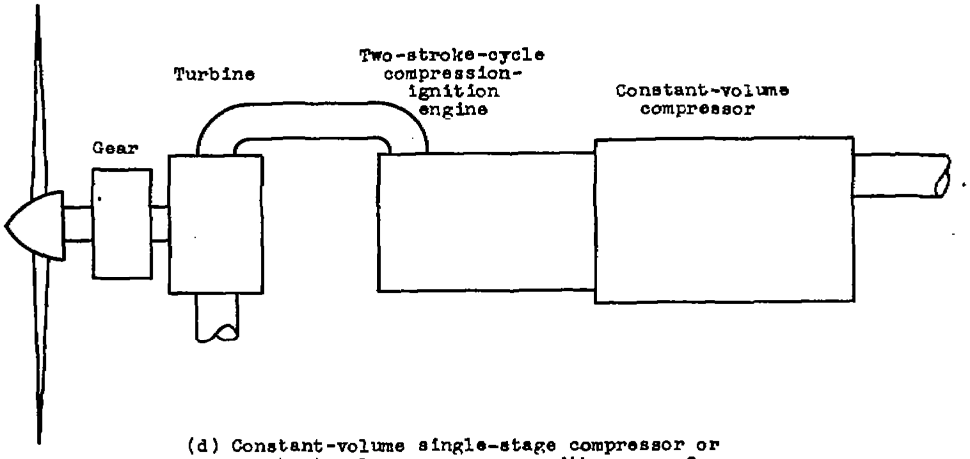
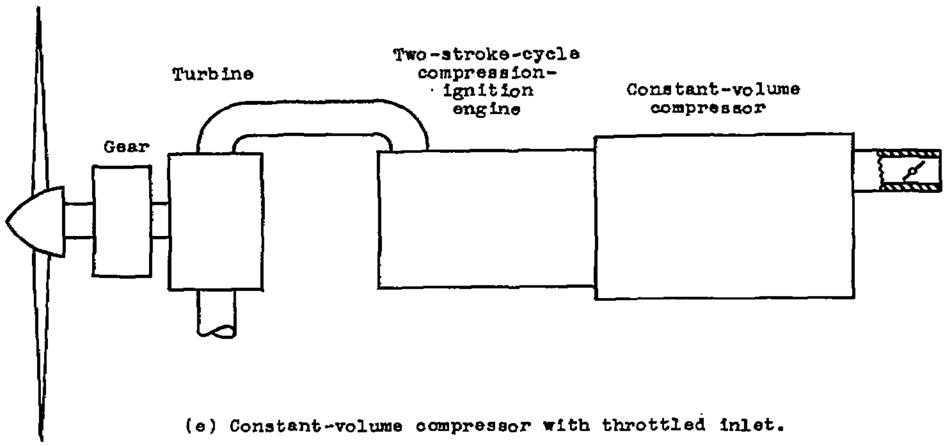
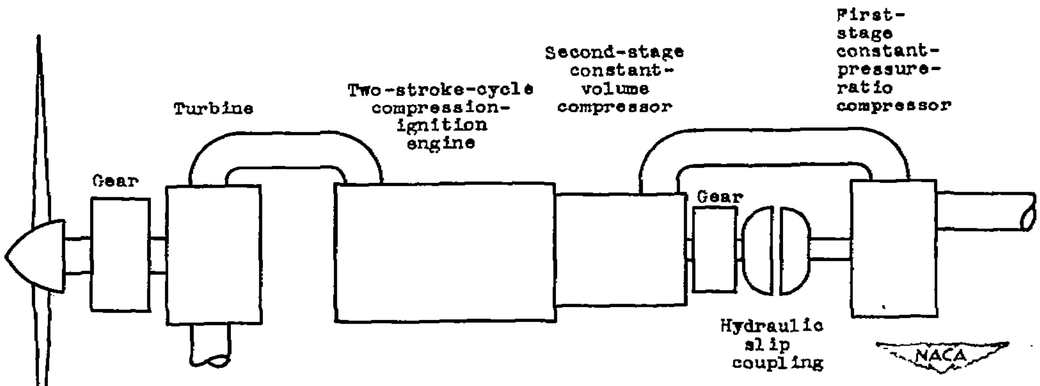


(d) Constant-volume single-stage compressor or constant-volume compressor with means of varying the volumetric capacity.



(e) Constant-volume compressor with throttled inlet.



(f) Multistage compressor comprising constant-pressure-ratio first-stage compressor driven by hydraulic slip coupling followed by final-stage constant-volume compressor.

Figure 2. - Concluded. Schematic diagrams of systems used in analysis.

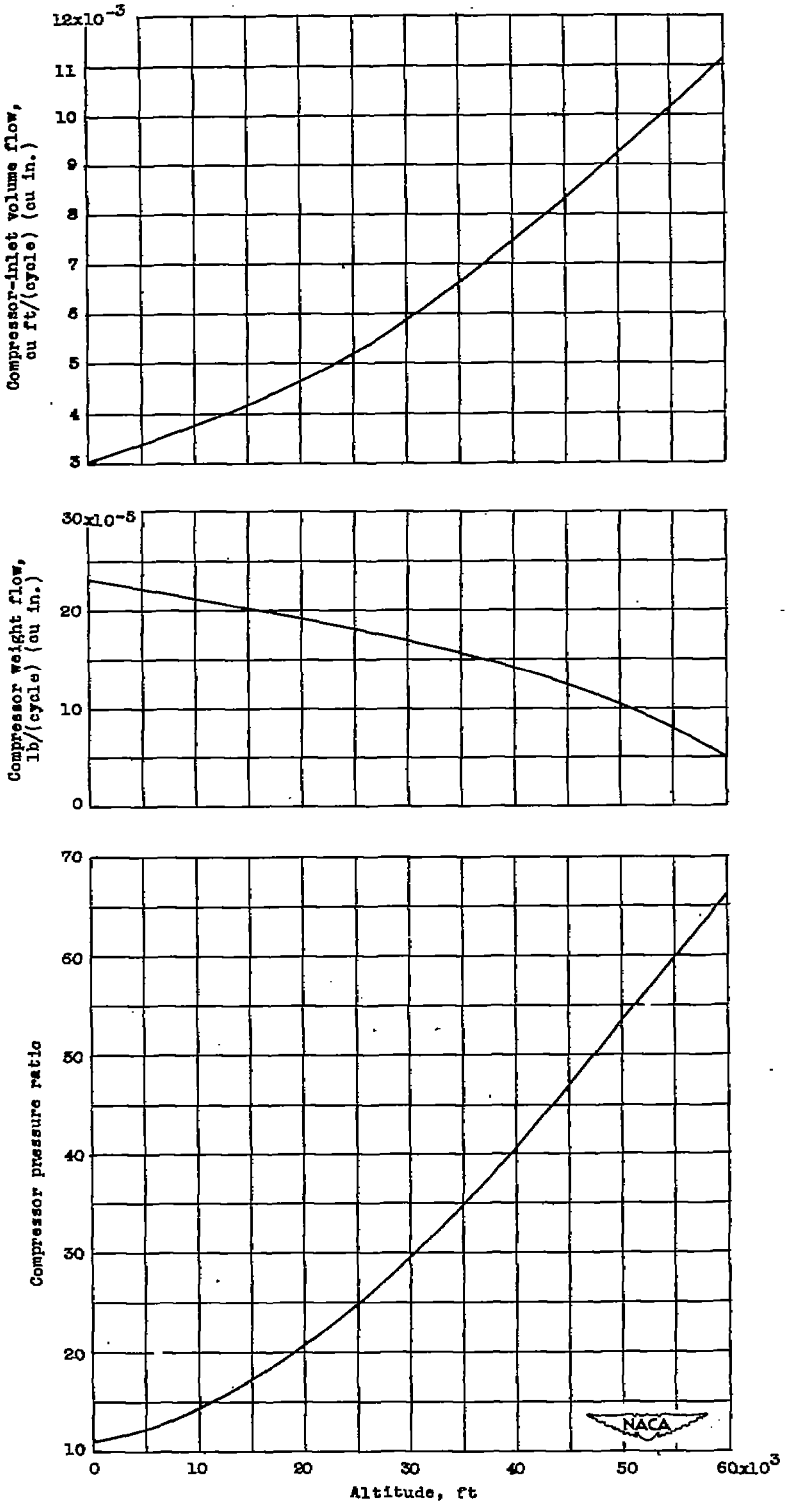


Figure 3. - Compressor requirements of ideal gas-generator engine. Scavenge ratio, 1.0.

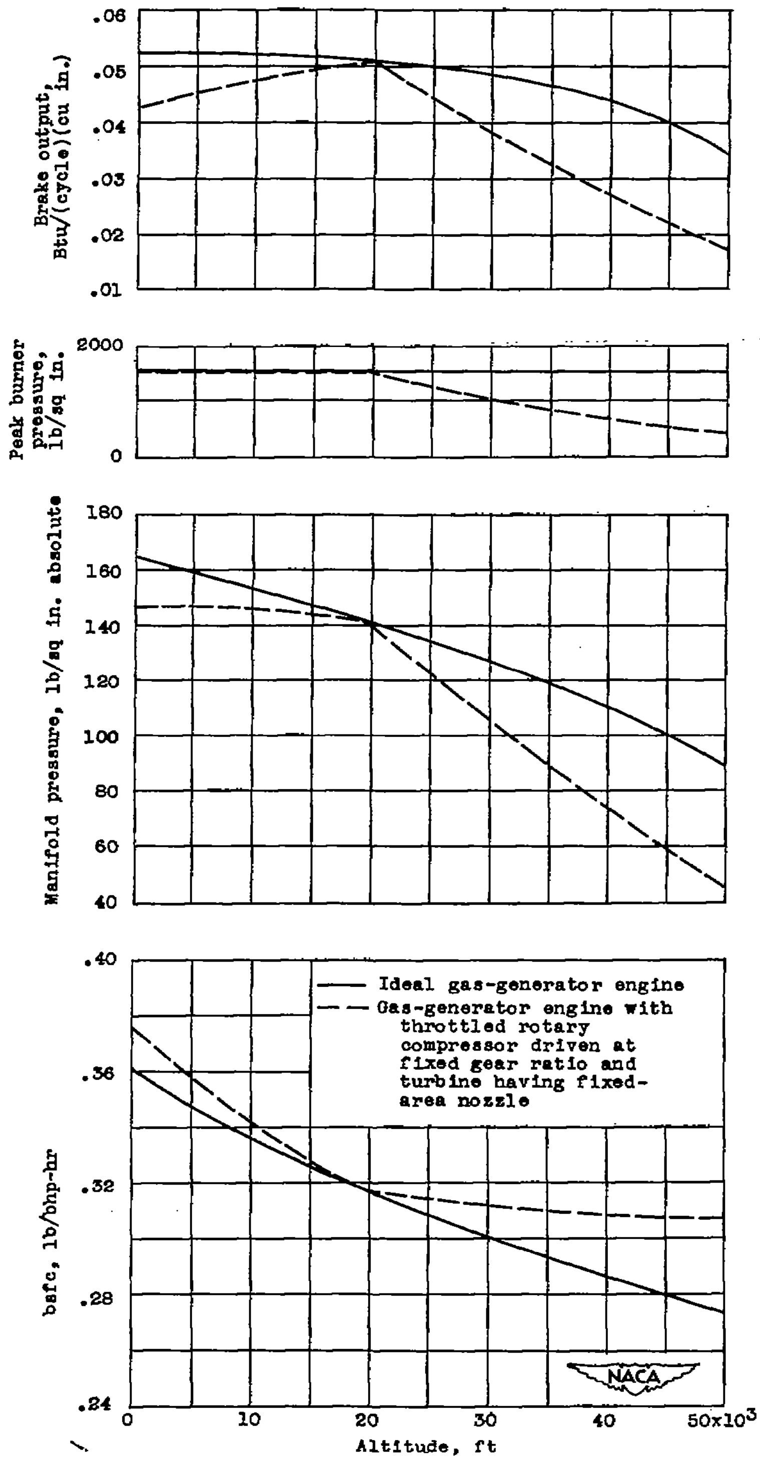


Figure 4.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating throttled rotary compressor driven at fixed gear ratio and turbine having f'xed-area nozzle.

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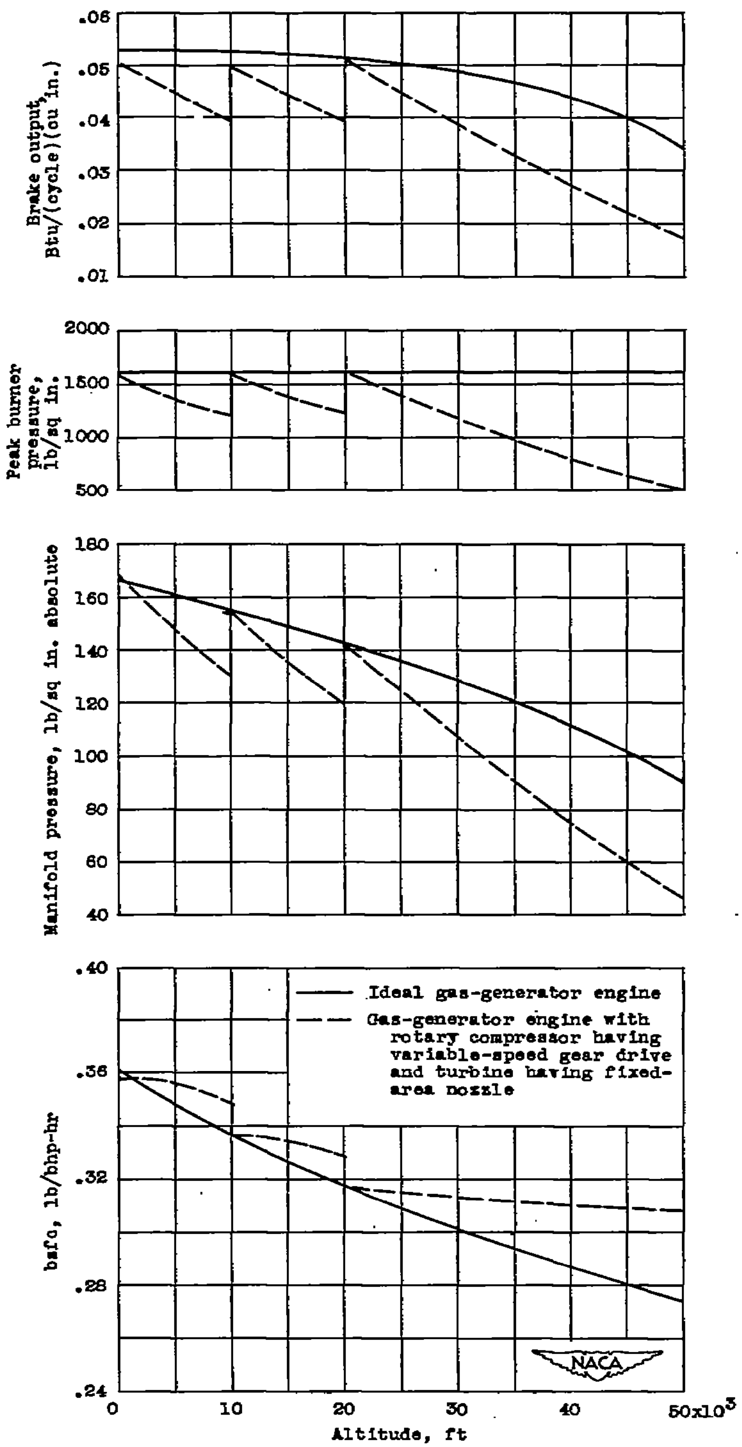

Figure 5.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating two-stage rotary compressor with first stage driven by three-speed gear and turbine having fixed-area nozzle.

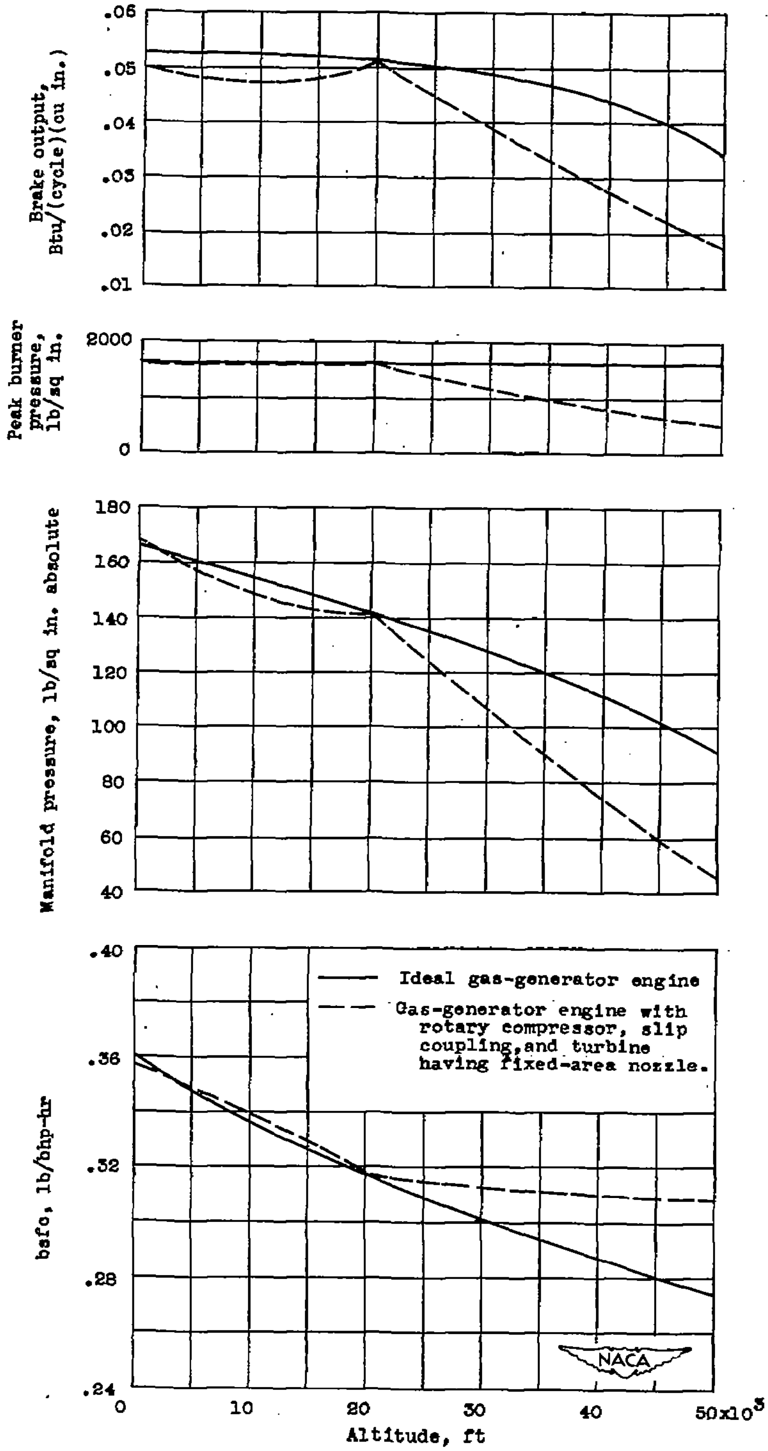

Figure 6.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating two-stage rotary compressor with first stage driven by slip coupling and turbine having fixed-area nozzle.

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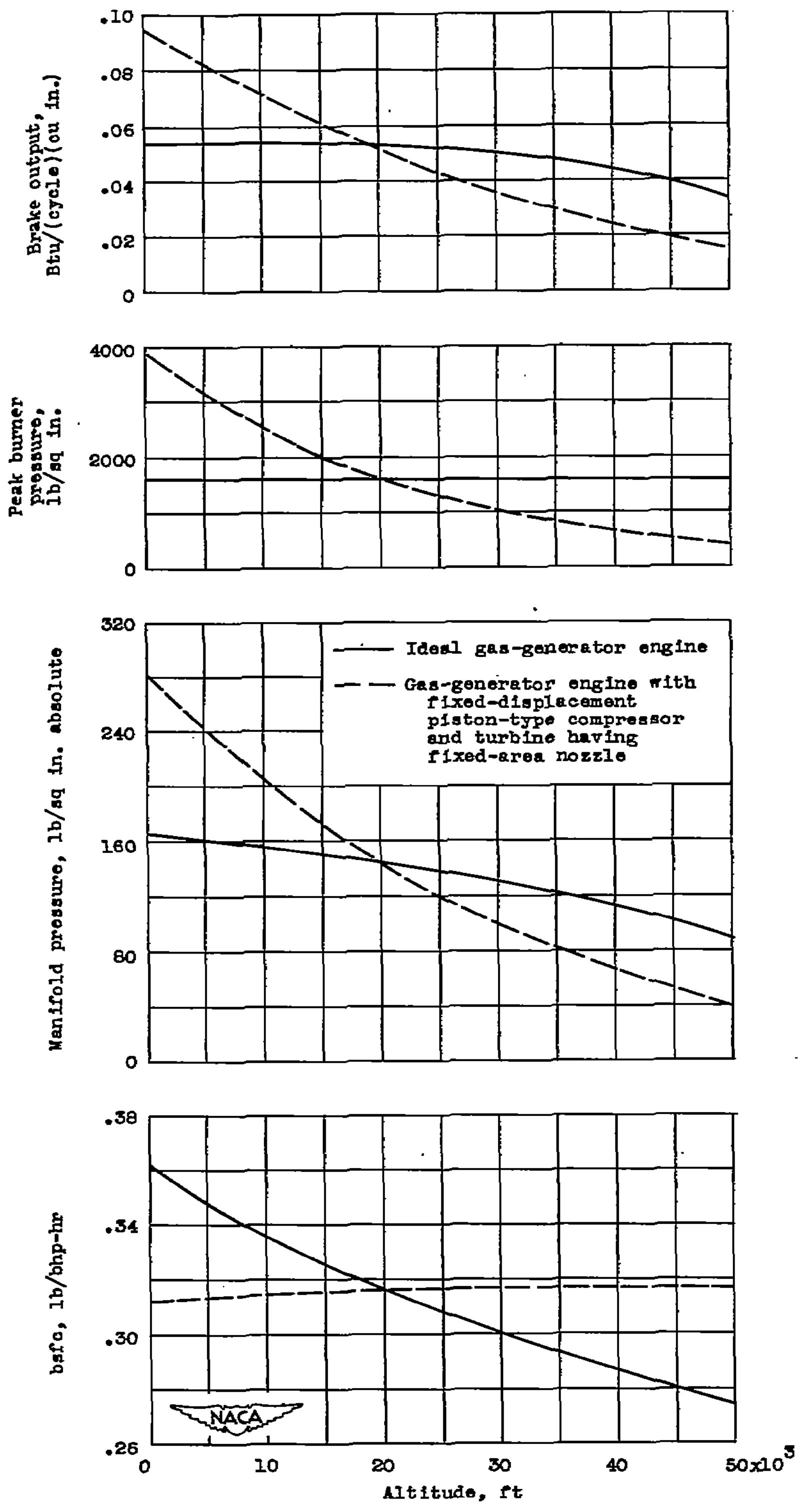

Figure 7.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating fixed-displacement piston-type compressor and turbine having fixed-area nozzle.

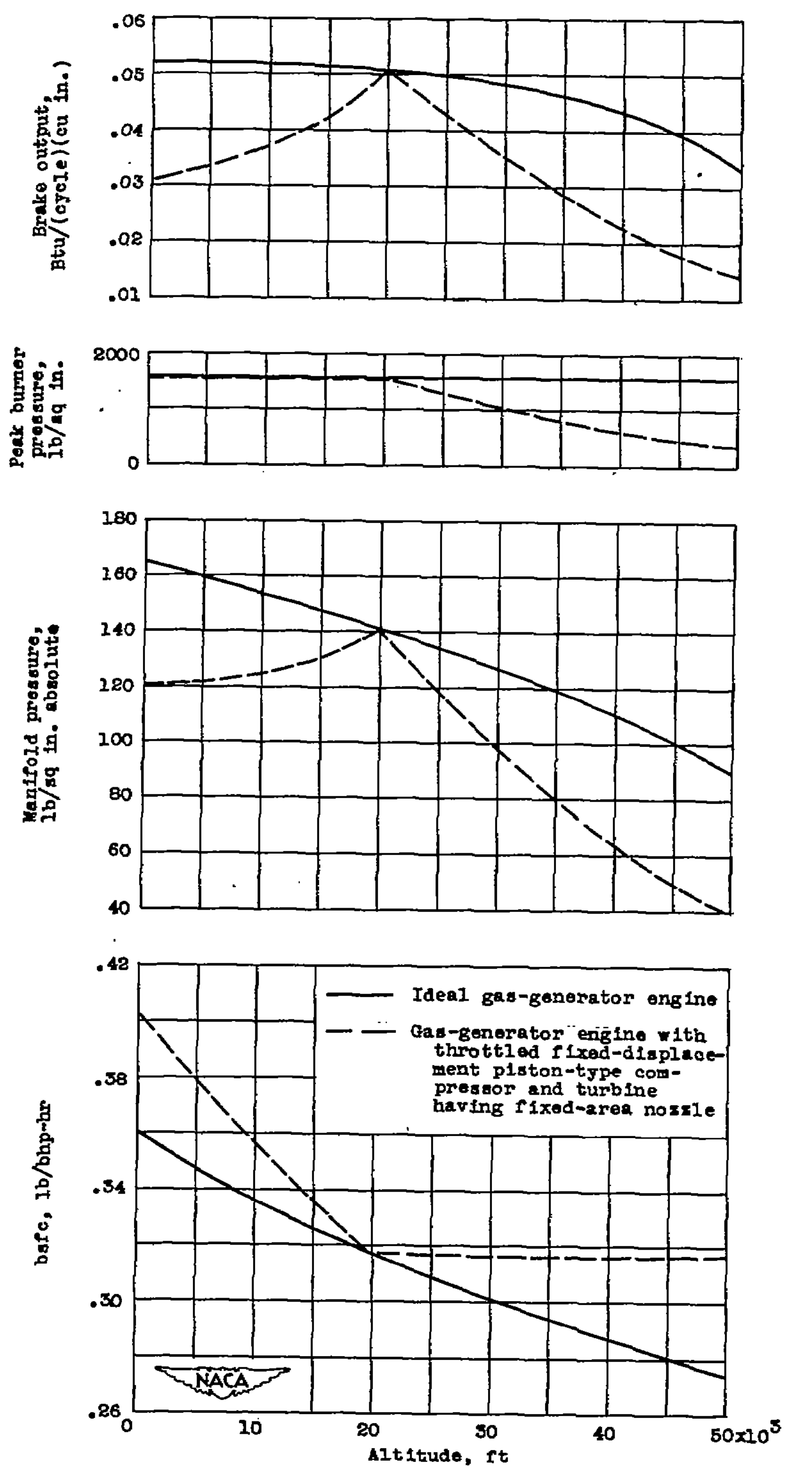

Figure 8.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating throttled fixed-displacement piston-type compressor and turbine having fixed-area nozzle.

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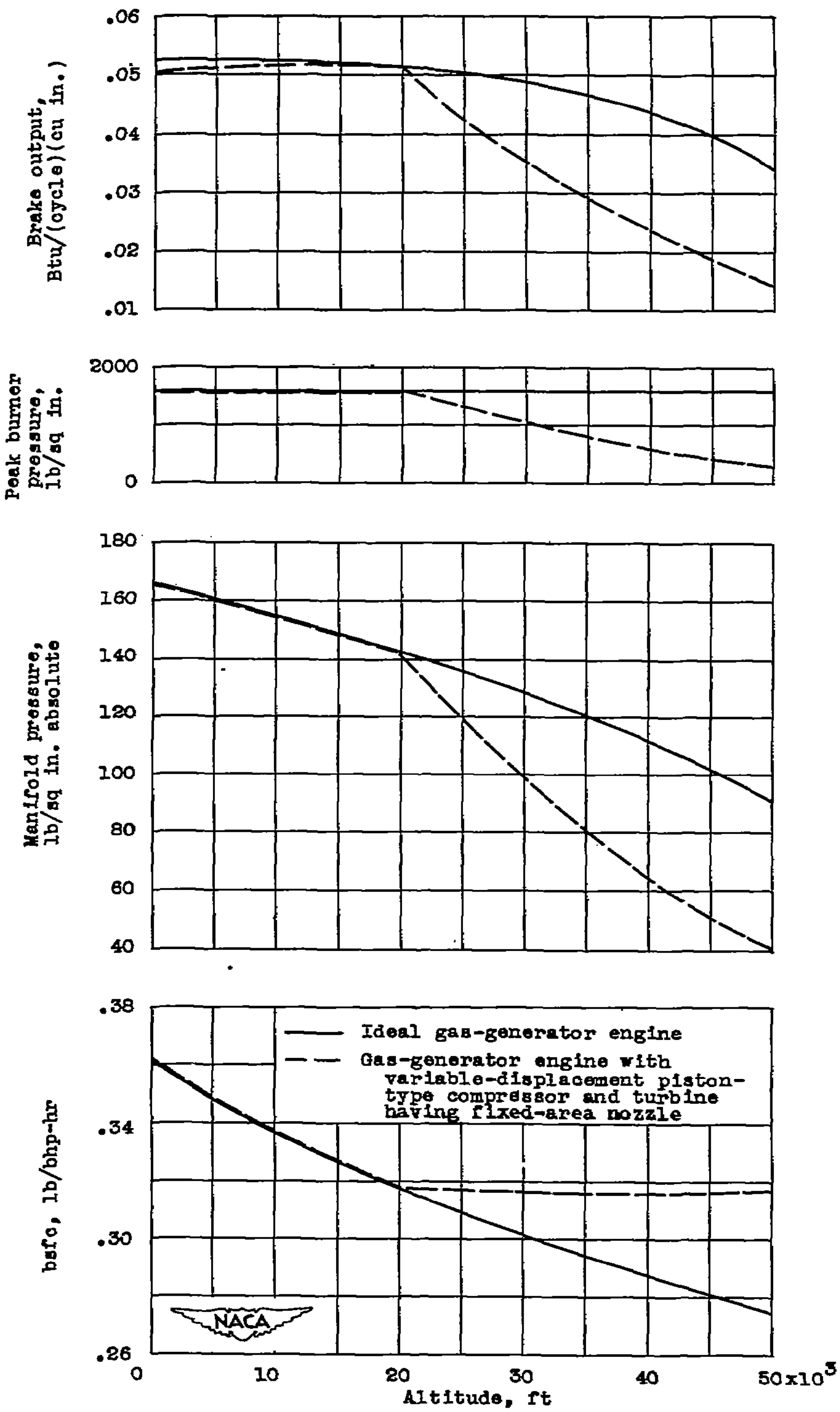

Figure 9.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating variable-displacement piston-type compressor and turbine having fixed-area nozzle.

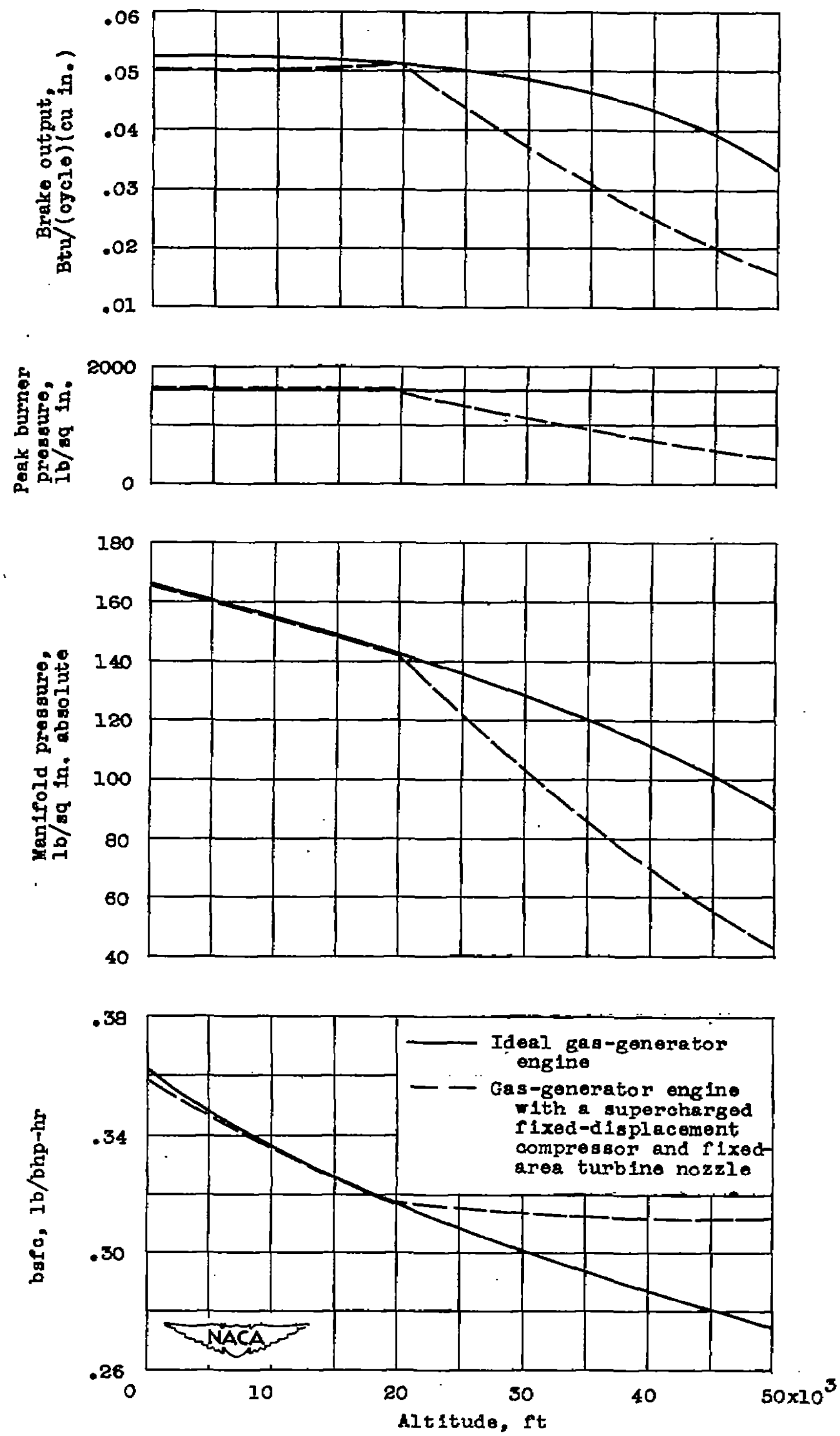

Figure 10.- Comparison of performance of ideal gas-generator engine with that of gas-generator engine incorporating fixed-displacement compressor, a supercharger driven by slip coupling, and turbine having fixed-area nozzle.

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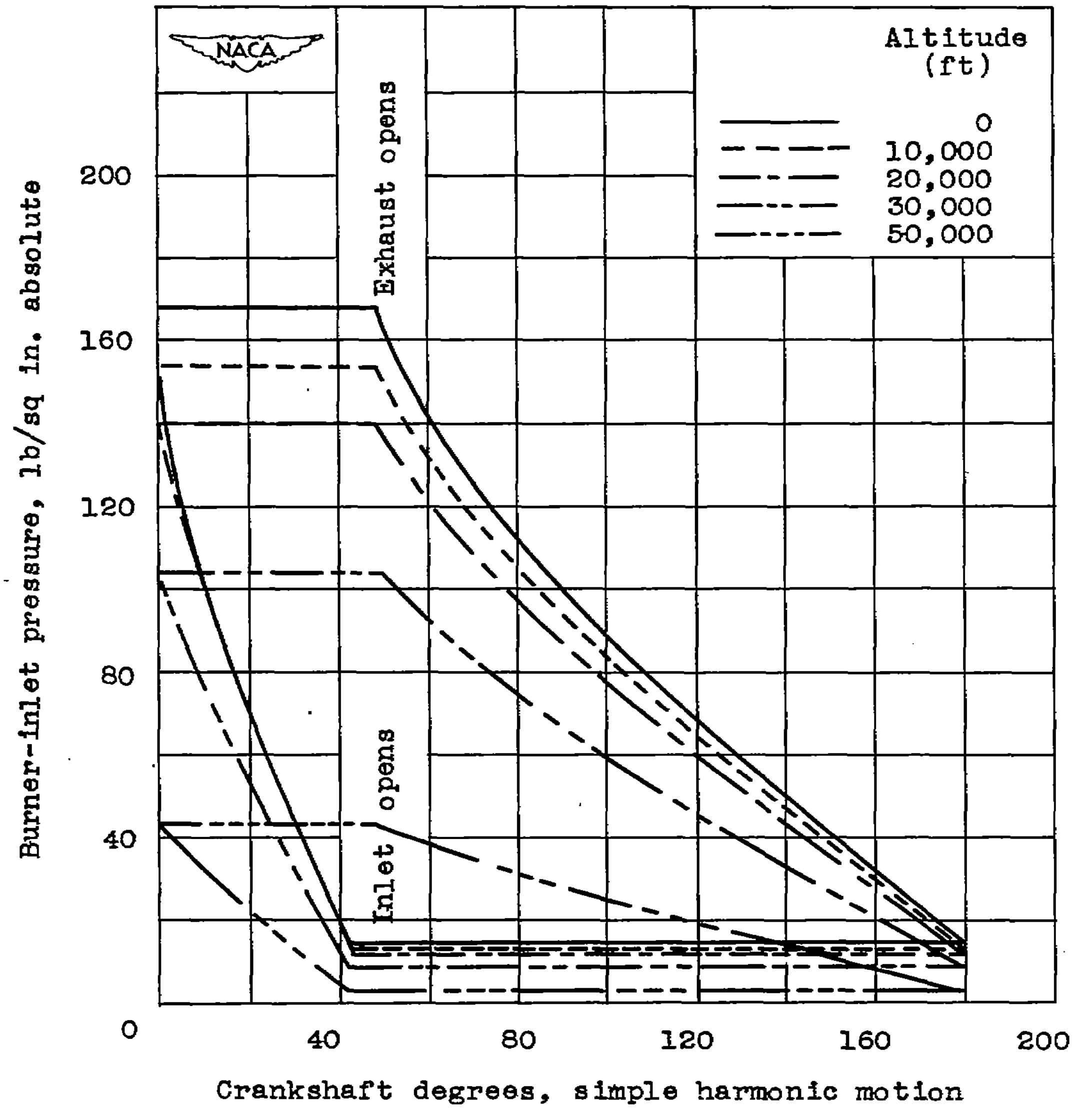

Figure 11.- Effect of altitude on indicator diagram of supercharged fixed-displacement piston-type compressor and turbine having fixed-area nozzle.

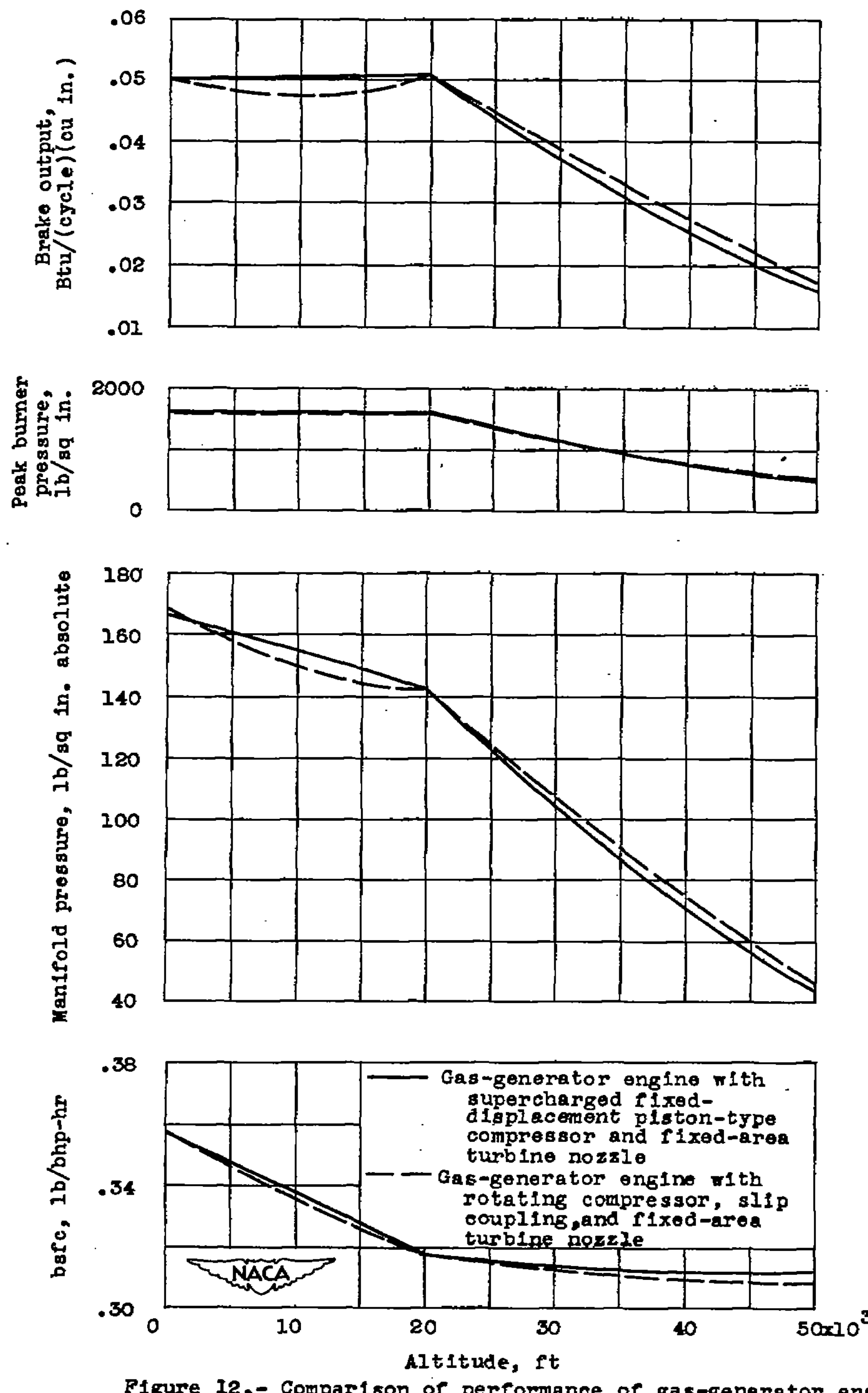

Figure 12.- Comparison of performance of gas-generator engine incorporating fixed-displacement piston-type compressor, supercharger driven by slip coupling, and fixed-area turbine nozzle with performance of gas-generator engine having two-stage rotary compressor with first stage driven by slip coupling and turbine having fixed-area nozzle.

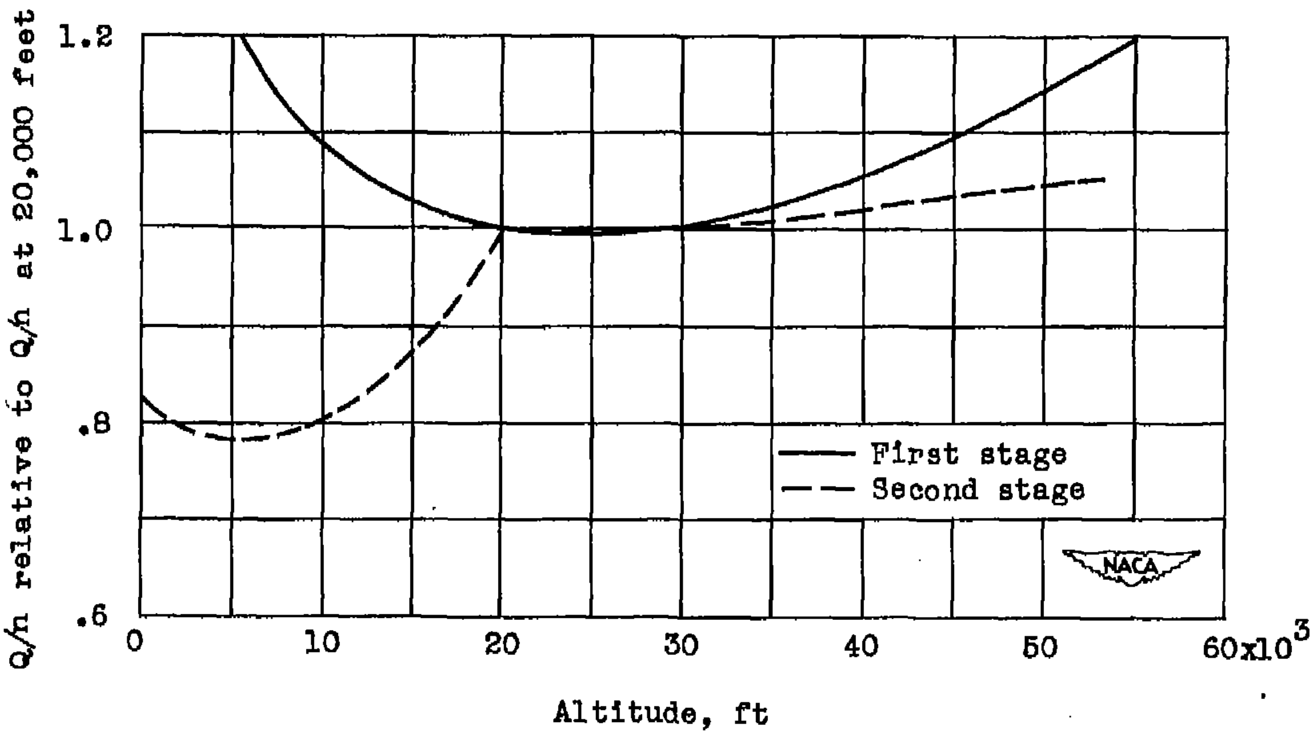

Figure 13.- Load coefficient Q/n at various altitudes relative to load coefficient Q/n at 20,000 feet for multistage rotary compressor with first stage driven by hydraulic slip coupling and turbine having fixed-area nozzle.

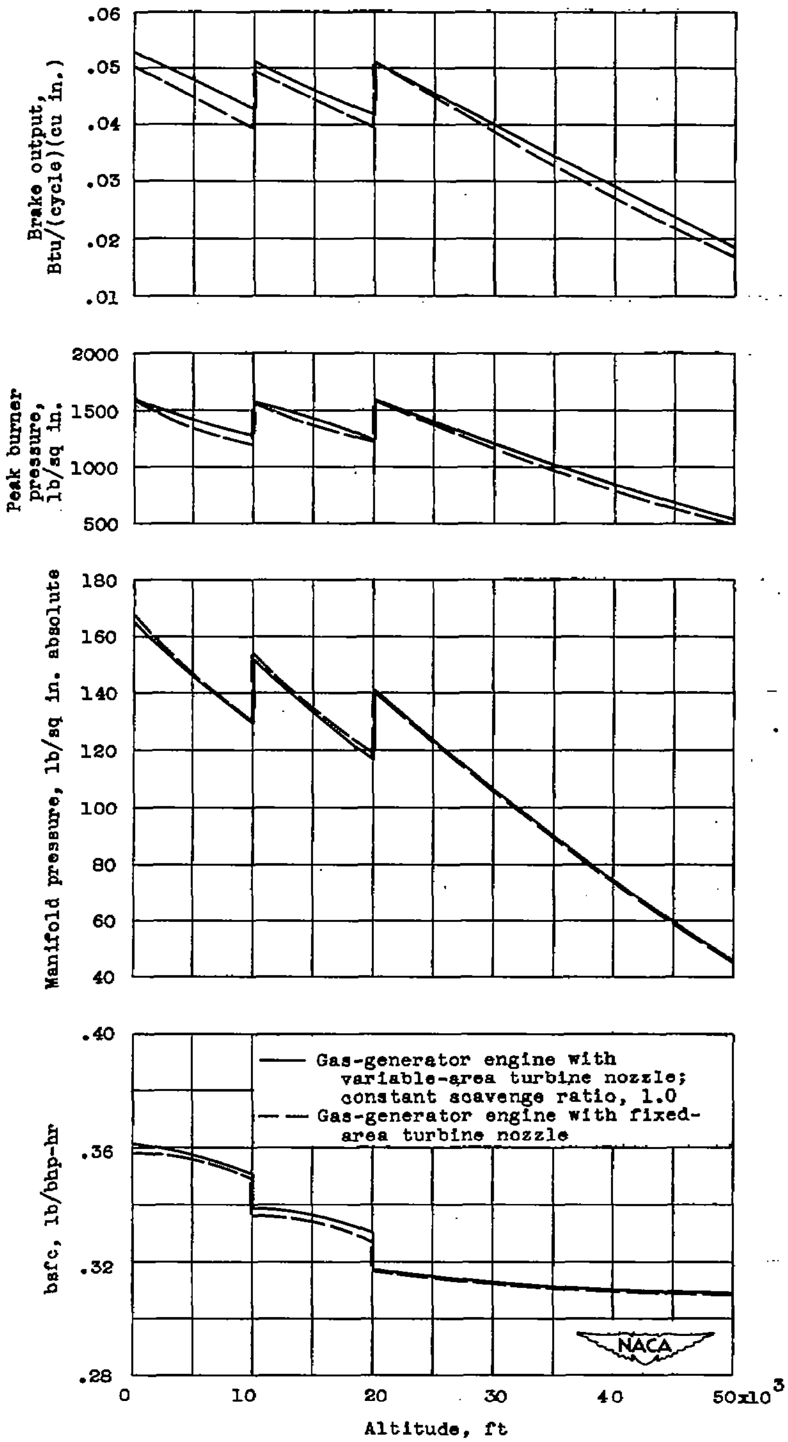

Figure 14.- Effect of fixed- and variable-area turbine nozzles on performance of gas-generator engines incorporating two-stage rotary compressor with first stage driven by a three-speed gear.

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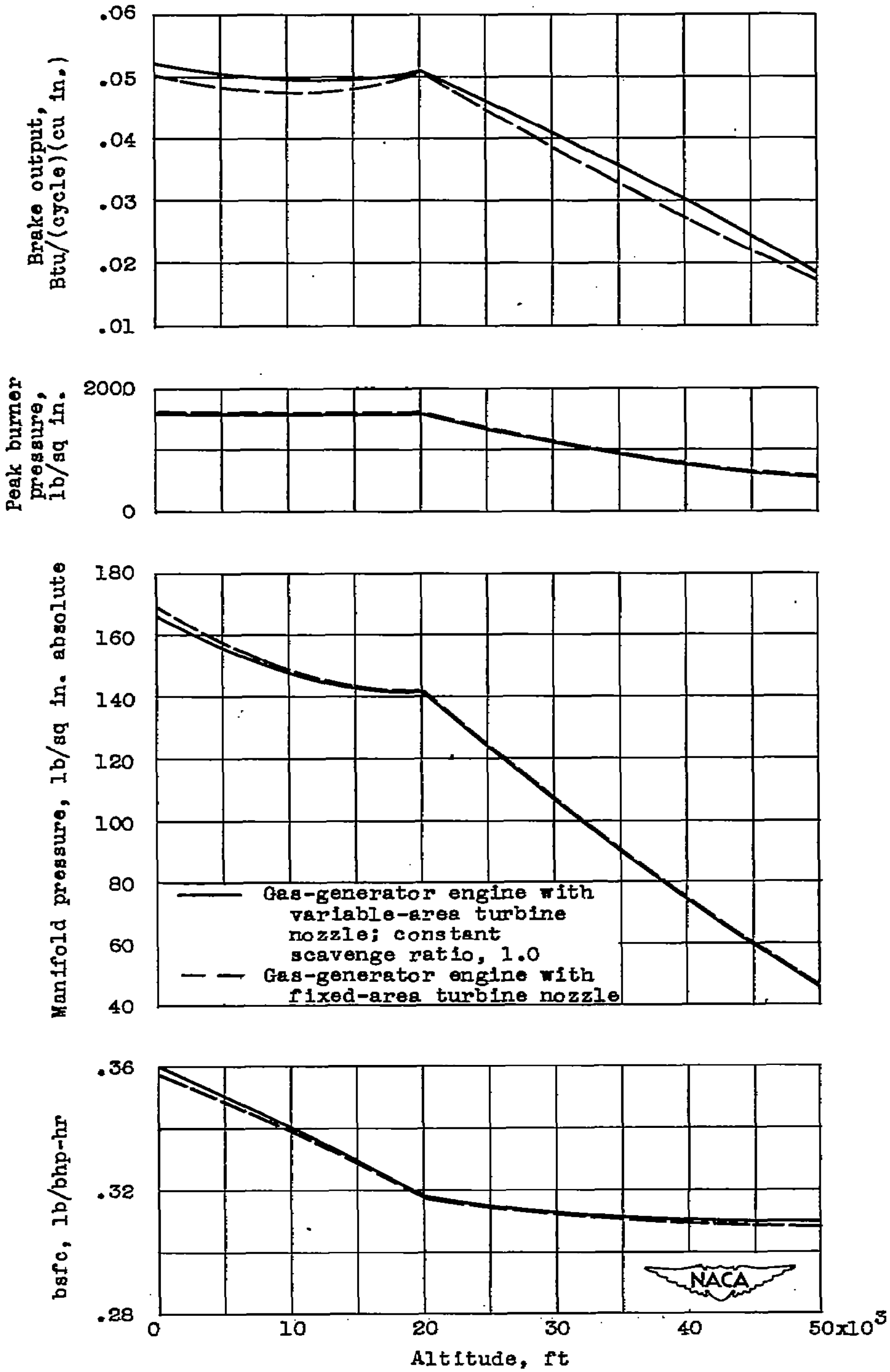

Figure 15.- Effect of fixed- and variable-area turbine nozzles on performance of gas-generator engines incorporating two-stage rotary compressor with first stage driven by slip coupling.

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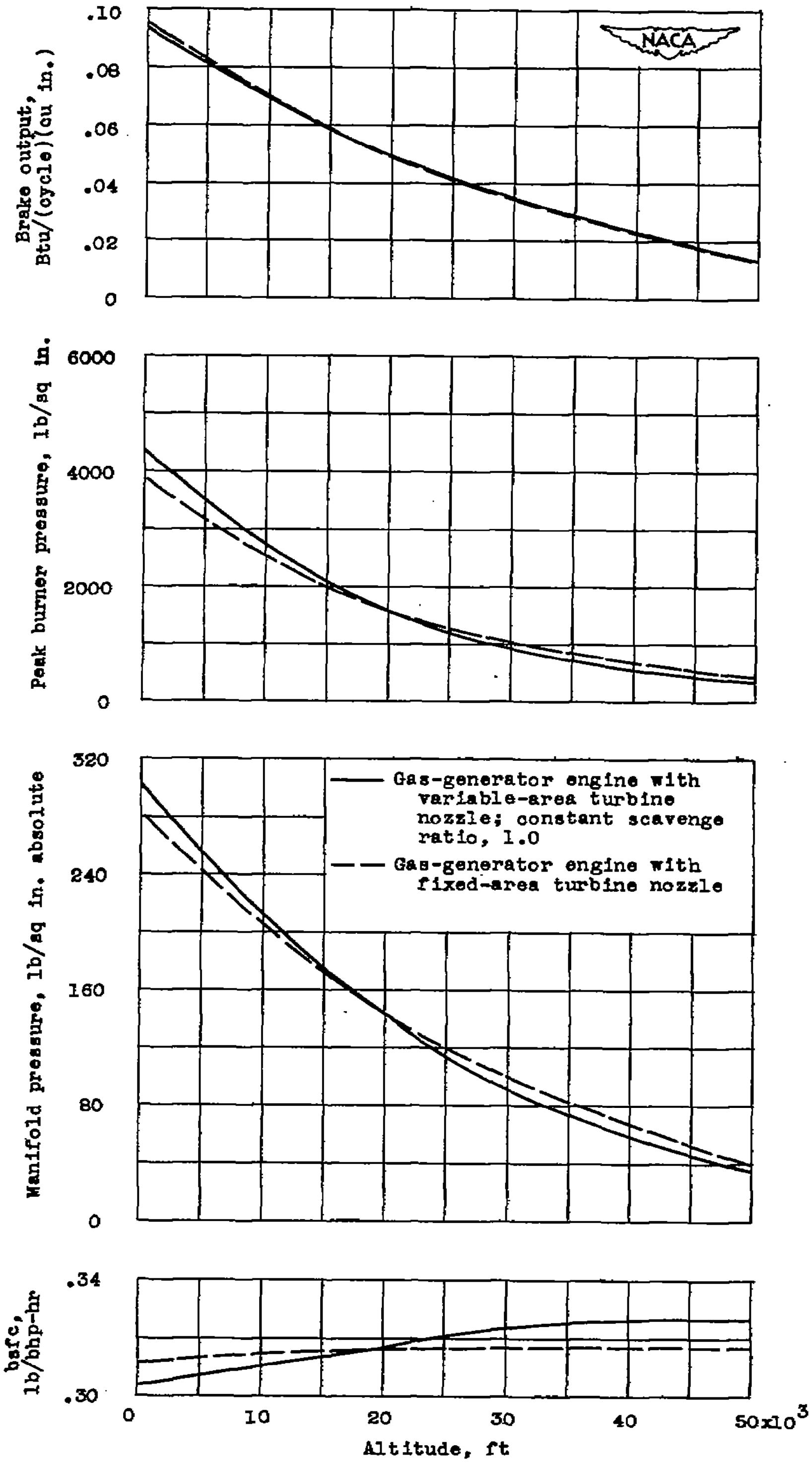

Figure 16.- Effect of fixed- and variable-area turbine nozzles on performance of gas-generator engines incorporating fixed-displacement piston-type compressors.

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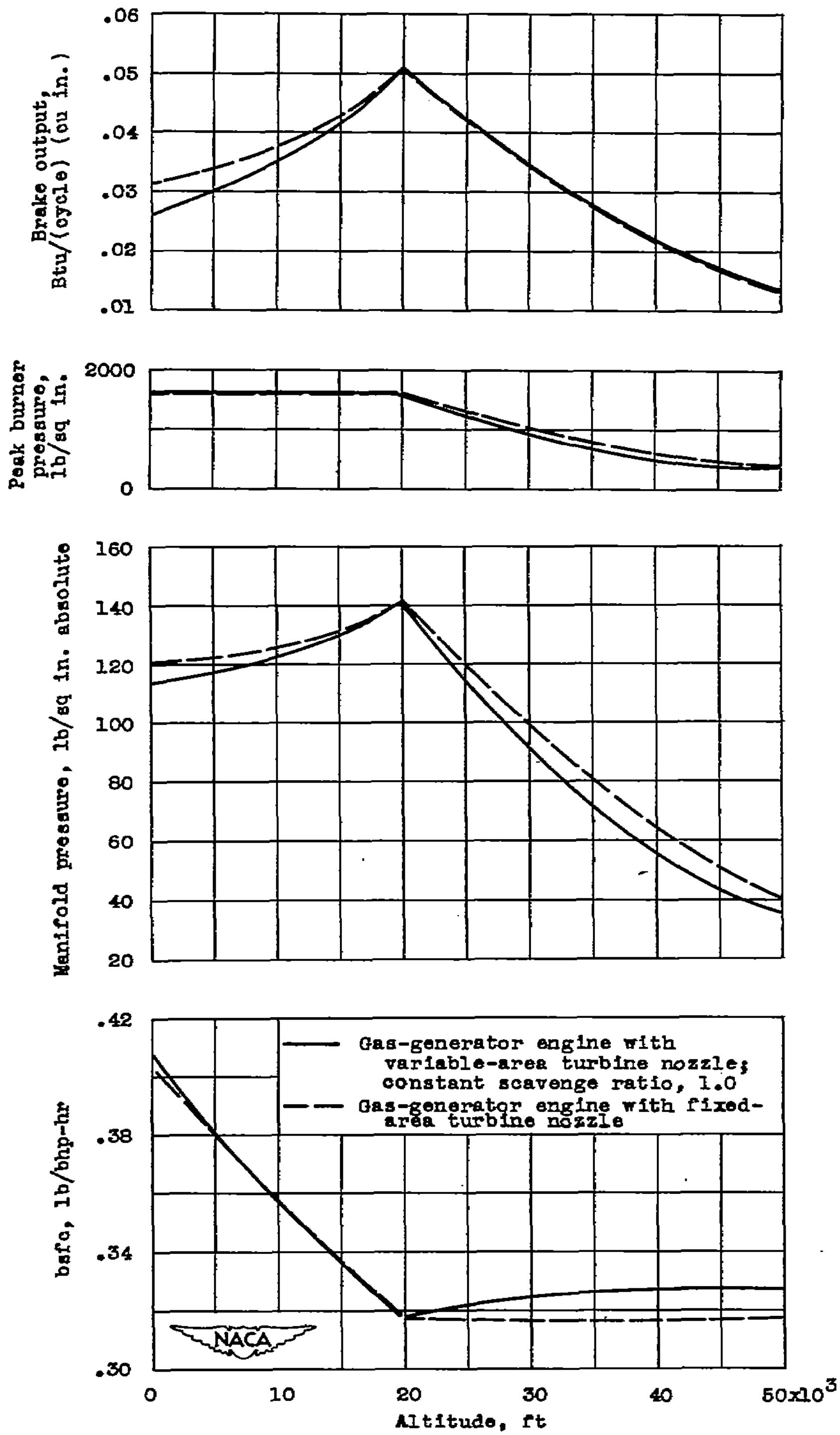

Figure 17. - Effect of fixed- and variable-area turbine nozzles on performance of gas-generator engines incorporating throttled fixed-displacement piston-type compressors.

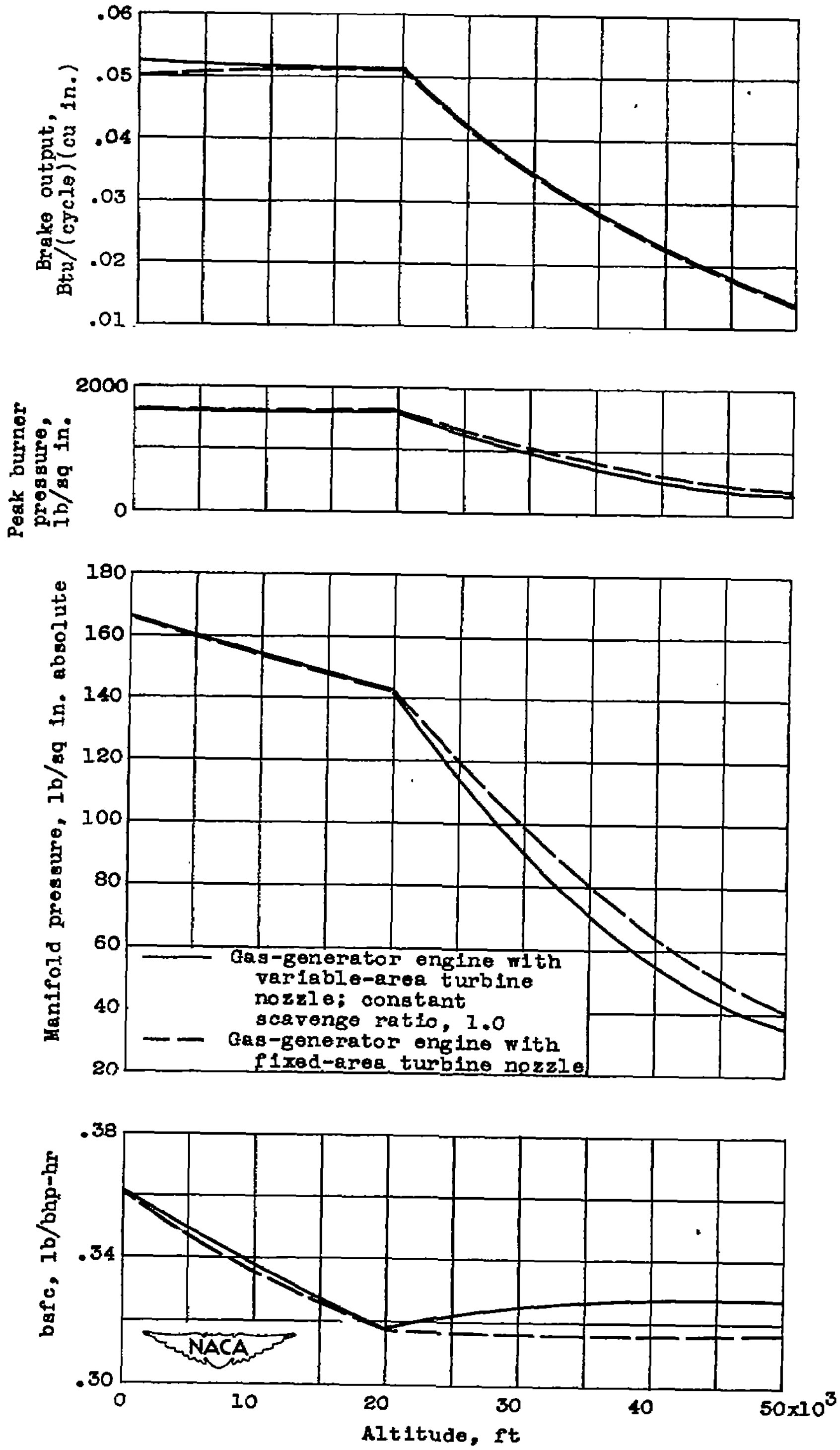

Figure 18. - Effect of fixed- and variable-area turbine nozzles on performance of gas-generator engines incorporating variable-displacement piston-type compressors.

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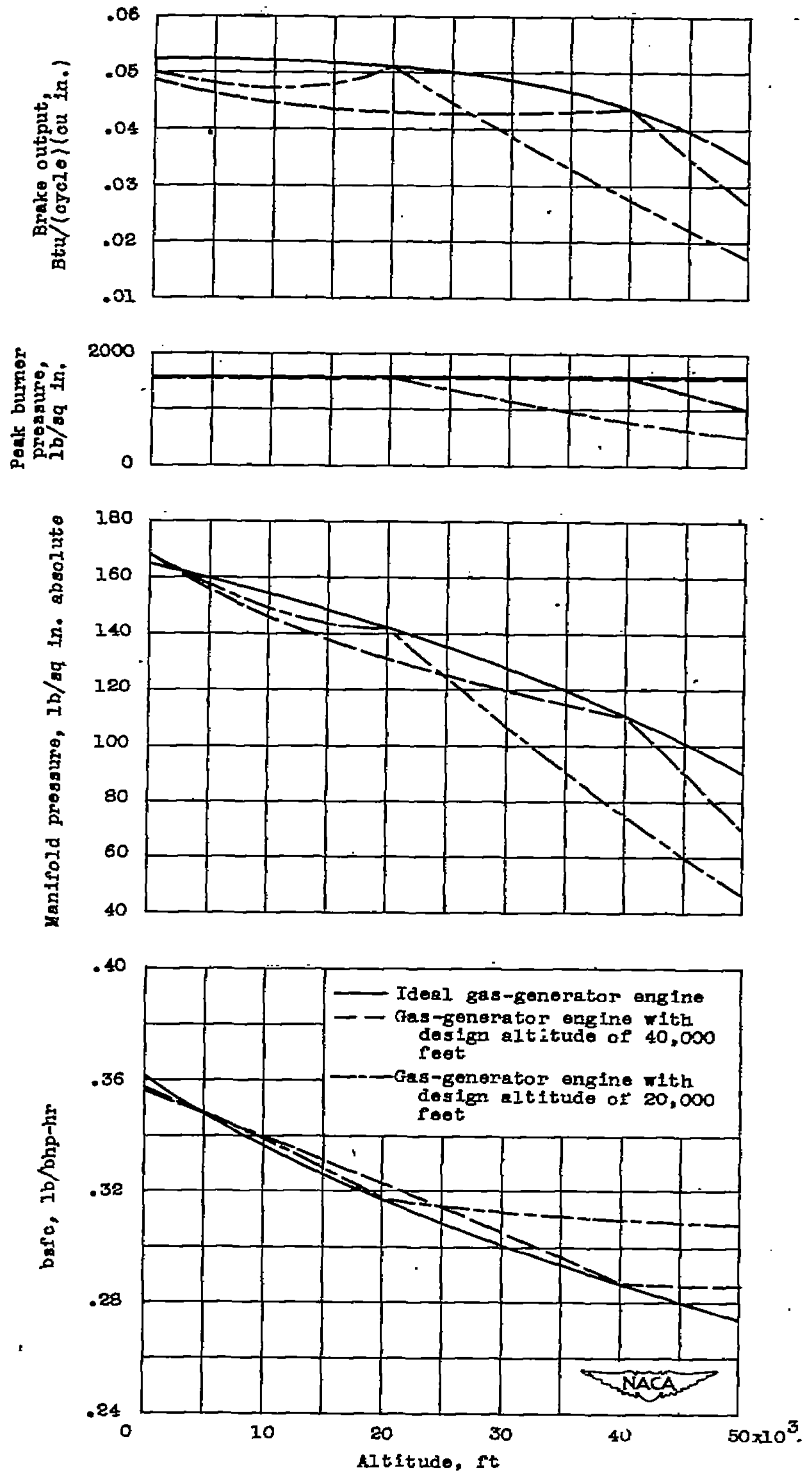

Figure 19.- Effect of design altitude on performance of gas-generator engines incorporating two-stage compressor with first stage driven by slip coupling and turbine having fixed-area nozzle.

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